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AIR CONDITIONING AFLOAT.

By Daniel S. Berres, Ernest A. Redman, George P. Lively,*.

Navy Practice Prior to 1940.

Mechanical cooling of ventilation air has been employed to varying degrees in surface ships of the Navy for more than 40 years. Until quite recently, the principal functions of such cooling were:

(a) To improve water tight integrity. Vital spaces below decks within armor were cooled to eliminate large ventilation ducts which would be open during battle.

(b) To maintain temperature of critical material such as ammunition. The location of magazines contiguous to heat producing spaces requires some positive form of heat removal and since the securing of ventilation during battle is mandatory

*The Authors are engineers in the Bureau of Ships, Navy Department.

by reason of damage control considerations, magazine cooling was employed.

(c) To limit the maximum temperatures in highly specialized compartments, such as ready rooms in aircraft carriers.

Practically all other compartments except storerooms not requiring treatment were ventilated. Ventilation was designed to limit the dry bulb temperature rise over weather air temperatures in varying degrees depending upon compartment use. Ventilation of machinery spaces was primarily designed to provide cool oases at watch stations and also to limit the temperature rise in the overhead.

Heating and cooling equipment had not been standardized. Fans, coils and compressor condenser equipment while purchased under rigid specifications, were in fact, commercial equipment modified to meet such specifications. Subject to these specifications, vendors adapted their equipment to meet the performance requirements for a given vessel. Plans were processed before the desired type of equipment was fixed. Manufacturers could not possibly meet the requirements without oversizing some if not all equipment furnished. For example, if a specification required a 2315 cfm fan to develop 2.2″ of static pressure, it was highly improbable that any manufacturer could exactly meet that requirement. He would accordingly offer the nearest size of fan manufactured by his organization. The duct system would probably have been completely detailed by the time the contract was awarded, and the installation of the fan furnished would result in excess capacity in the system. Similar examples could be cited for cooling coils which resulted in variations of the type of heat transfer surface from each supplier. Tube diameter, tube spacing, fin type and spacing, and, of course, equipment performance varied with the different suppliers.

Mechanical cooling installations all employed the rugged medium speed type of Freon-12 compressors which had given satisfactory performance for refrigeration service. The same material specifications were used for both refrigeration and mechanical cooling compressor condenser equipment.

Compressors were limited to maximum speeds of 600 rpm. with suction valves in pistons of the trunk type. Rugged condensers designed especially to withstand the erosive and corrosive effects of sea water, were of the horizontal multipass type with

tubes and tube sheets of copper nickel and water boxes of gun metal or valve bronze. The design of these condensers restricted water velocity in the tube to 4 feet per second. Compressor condenser plants probably averaged less than 10 tons of cooling per plant. Under the foregoing conditions, not only were spare parts excessive, but also battle damage stocks were virtually impossible to maintain.

Results achieved by these installations were in general as expected. Extensive studies had revealed that in tropical waters injection temperatures at sea would be approximately 85° F. with weather air temperatures approximately 88° F. It was known that in berthing spaces, dry bulb temperatures might be 100° F. with effective temperatures of 87°. Effective temperature, (E.T.), a scale developed by the American Society of Heating and Ventilating Engineers, and adopted by the Navy Department, is an arbitrary index which combines into a single value, the effect of dry bulb temperature, humidity and air movement to reflect the sensation of warmth or cold felt by the human body. It was further known that such conditions would result in personnel discomfort. It was not known whether continued exposures to such conditions would result in a reduction of personnel efficiency. Further improvement of habitability was presumed to be a comfort item. Weight, space and power additions incident to such improvement were not considered compatible with Naval Warship design practice.

Wartime Changes in Design Practice.

By August 1942, ventilation design practice was modified to reduce the maximum acceptable dry bulb temperature rise over weather air in living quarters from 10° F. to 7° F. This practice in effect limited the maximum temperature in berthing spaces to 95° F. Unfortunately, the 7° F. design was the limit to which ventilation could be extended since such reduction required an increase of about 45 per cent in air quantities in a given space. Mechanical cooling in hospital spaces, was first installed in the U.S.S. *South Dakota*. By late 1942, war experience demonstrated the necessity for the installation of mechanical cooling in all operating rooms and sick bays of vessels larger than destroyers. This practice was gradually extended until the U.S.S. *Haven*, converted in 1945 from a tanker to a hospital ship was the first

Naval vessel of her type to have living quarters, as well as hospital spaces mechanically cooled. Weight and space considerations on this type of ship are of lesser importance than on a warship and comfort especially in medical spaces is of prime importance.

The extended use of high heat producing electronic equipment in new types of spaces, such as combat information center required that mechanical cooling be employed; top side control spaces such as flag plot had to be so cooled; photographic laboratories on carriers required similar treatment; increased heat loads in already cooled spaces required additional cooling machinery. These requirements in addition to the requirements for normally cooled spaces of the numerous vessels being built or converted necessitated the procurement of prodigious amounts of equipment.

Equipment Development.

At one time a number of self contained units of standard commercial design, but with special water cooled condensers for use with sea water were installed. These units were, however, generally unsatisfactory and had to be replaced.

As the war progressed, procurement of equipment designed for a definite service became impossible within the time alloted to make the installations. Many Maritime Commission vessels were taken over, converted for Naval use and put into service in less time than the normal procurement period for refrigerating equipment and the necessary motors and controllers.

To meet these requirements, purchase specifications were developed to cover standard plants of 1.8, 3.8, 6.2, 11.0 and 17.0 ton capacities. With each size plant was purchased electrical equipment of three standard current characteristics, 440 volt 3 phase 60 cycle, 115 volt D.C. and 230 volt D.C. All equipment on these various contracts was strictly in accordance with Navy specifications.

The use of these standard "high sides" was restricted as far as possible to conversions and to vessels requiring supplementary installations to those covered by the original design and only in vital spaces. For new construction, it was generally possible to secure deliveries of equipment to the building yard in ample time for orderly installation by placing orders during the early stages of construction.

While this system insured the installation of the most rugged and substantial equipment obtainable, the great number of compressor units often installed on larger vessels created a difficult maintenance and service problem. On one class of vessels, the original design involved the use of 17 Freon-12 compressors. Additional cooling requirements made necessary by added equipment, and by directives to cool additional spaces increased the number of compressors to well over fifty, so that the trained personnel available for servicing was unable to give satisfactory attention to the equipment.

Under such conditions, the weights of equipment involved also became excessive, and the design trend changed to require a smaller number of larger units to serve contiguous or closely grouped spaces.

During the later periods of the war, the cooling requirements for new vessels became more stabilized and it was possible to install seven or eight compressor units to secure far more air conditioning effect than had previously been obtained with a greater number of units, simultaneously decreasing weight, and simplifying maintenance.

These developments however, involved no changes in basic design of the high side equipment. Compressors and condensers were of the type that had proven their ability to perform satisfactorily under all conditions of service and weights were considered of secondary importance. It was considered highly undesirable to sacrifice reliability to any other consideration and the type of equipment being installed had demonstrated this reliability. Chances could not be taken with experimental or unproved equipment when every vessel was required to be at the peak of efficiency at all times.

Recent studies and experimental purchases leading to the evaluation of lighter Freon compressor condenser equipment and other types of refrigerating machinery peculiarly adapted to mechanical cooling of air aboard ship have been undertaken. Three types of equipment are under study. The first consists of a recently developed high speed Freon-12 compressor, by the use of which the same refrigerating effect can be obtained in a much smaller, lighter package than the conventional compressor. For use in connection with these compressors, investigations are to be made of a condenser having finned tubes, but with materials

found most suitable for use with sea water. Specification requirements have been somewhat modified to promote additional weight and space economies. Cooling throughout the system by direct expansion is contemplated. The second is the conventional steam jet system with minor modifications found desirable for Naval service, and designed with primary consideration being given to refinements for weight saving and reductions of space occupied. Chilled water will be circulated as the refrigerant in a closed system. Space and weight requirements for the refrigerating unit in combination with its low side equipment are competitive with those of the direct expansion units. The maintenance problem is expected to be simpler than with Freon compressors.

The third system is not yet in a form that the designers consider suitable for Naval installations, but experimental and commercial applications indicate that it may have valuable possibilities. It is an absorption system in which chilled water is produced, then circulated in a closed system to cooling coils located as found desirable for proper distribution of refrigerating effect. Preliminary studies indicate that weights are about the same as for steam-jet units of equal capacity, but that space requirements are definitely less. Steam consumption, appreciably less than for the steam jet plant is expected.

Immediately prior to and during World War II, a series of Navy Standard fans was established which greatly reduced the number of different sizes of fans. This series consisted of 16 axial fans and 9 centrifugal fans which were expected to cover at least 90 per cent of all the Navy's requirements. Fan performance curves of all interested manufactures were carefully studied to determine which commercially available fans could best be fitted into a Navy series. A family of fans was selected and established as Navy standard. Principal overall dimensions such as length, bolt spacing, etc., and performance were rigidly fixed. Contracts were let with major fan manufacturers and plant capacities were concentrated on these restricted sizes. Battle damage replacement stocks were established and new ship designs utilized standardized fans with a resulting ultimate saving in production and repair time, and in equipment cost.

Ventilation heaters were similarly standardized. A review of existing copper tube stock and fabrication equipment was made

to first determine that type of surface which could be readily obtained. Straight plate fins were used primarily to facilitate cleaning, and by the middle of 1942, 14 sizes were established to make up the Navy standard heater group. These heaters were so dimensioned that, by proper selection, heaters could be picked to meet any reasonable performance and space requirement. Early application of these heaters proved that complete coverage necessitated more than one fin spacing. Consequently, by the middle of 1943, three standard fin spacings, namely, 3, 5, and 7½ fins to the inch were established. Selection of heaters to do a specific job by the then current commercial procedure was tedious and time consuming. By January of 1944, a selection chart greatly simplifying selection procedures was devised and issued by BuShips. Selection time was reduced approximately 80 per cent and the possibility of arithmetical error in selection was removed. A complete line of temperature regulators for heaters was simultaneously developed, utilizing four basic models and various poppet sizes covering a capacity range from 10 to 2700 pounds of condensate per hour. One of the models, incorporating two poppets in a single valve body is utilized as nonfreeze protection for preheaters, assuring steam flow during freezing weather. A Navy standard convection heater was also developed and is supplied in 10 sizes ranging from 1280 Btu/hr. to 20,000 Btu/hr. It is interesting to note that on one cruiser, the space occupied by an old type bare pipe heater and bypass was converted into a chief petty officer's pantry incident to change to the new finned heater.

A slide rule type heater selector was issued early in 1944 replacing the selector chart and incorporating sufficient information for selection of temperature regulators and convection heaters, as well as duct type heaters.

Initiation of work toward cooling coil standardization is represented in experiments conducted by the Engineering Experiment Station at Annapolis, Md., in which it was determined that a minimum fin thickness of 0.014″ was required to withstand shock due to gun fire. The American Society of Heating and Ventilating Engineers was requested to fully develop specifications for a U. S. Navy coil on a then outstanding contract. Actual test work on this contract was subcontracted to the Case School of Applied Science. After the initial work

was completed, direct contracts with Case were let to continue the work. Actual testing was begun in July of 1942 and is continuing.

It was the primary object of these investigations to establish a specification for a coil to obtain minimum weight, space and power requirements, minimum water carry over characteristics, ruggedness to withstand Navy use, maximum cleanability, ability to withstand shock, and to determine the effect of pitch and roll on performance. A common performance specification, incorporating approved overall dimensions, was written, on which coils were purchased from eight interested manufacturers. Each of these coils was thoroughly tested for capacity, loaded with air borne dirt, cleaned and rechecked. Some of the problems encountered in fields other than heat and thermodynamics were unique, for example a standard Navy dirt was determined, a procedure for accelerated dirt tests was developed and a pitch and roll machine was built. Two complete specifications were proposed, one for continuous plate-fin coils and one for integral round fin coils, as it was expected that the continuous plate fin would be used on surface vessels and the more costly but more efficient (greater heat removal per square foot of coil) integral fin coil be used in submarines where space is much more critical. The established plate fin requirements were briefly as follows:—

a. Tubes to be $\frac{5}{8}$″ O.D. copper and 0.035″ minimum thickness.

b. Horizontal tubes to be spaced in vertical rows with successive rows forming staggered arrangement of tubes in direction of air flow.

c. Tubes to be spaced on $1\frac{1}{2}$″ centers both horizontally and vertically.

d. Fins to be continuous and to project not more than $\frac{3}{16}$″ from extreme outer face of tube on both entering and leaving faces.

e. Fins to be smooth, 0.016″ thick, spaced uniformly, 7 fins per inch.

f. Fin sheets to have an integral fin collar to surround each tube. Tubes to be expanded into collars to form a tight bond.

g. Circuit length to be such that Freon pressure drop in each circuit be approximately equal and be between three and five pounds per square inch.

h. Tube circuits to be arranged for counterflow between Freon and air.

Tests showed that pitch of 7° or roll of 20° or combinations of both had no material effect on coil performance.

The tests also showed that the specification type coils can be readily cleaned and that coil capacity is little affected by dirt so long as air velocity is maintained. It is true, however, that dirt will materially reduce air velocity and procedures were developed for cleaning the coils.

A similar specification was prepared for integral fin type surface.

Some deviation from these specifications became necessary in the development of coils for shipboard use. While every effort was made to obtain counterflow, it was decided to sacrifice some efficiency to obtain a universal coil, that is a coil which can be used equally well with air flow in either direction. Further, although it was realized that a minimum number of rows is desirable for a given application, a standard depth of eight rows was selected. Navy application covers an extreme range of requirements, some of which could have been satisfied by coils varying in depth from 3 to 8 rows. However, considerations of simplification of procurement, replacement stocks, equipment selection, improbable fine control of superheat, and the numerous indications of higher latent heat loads than calculated all dictated the choice of the eight row depth.

Three coils were then built in accordance with the developed specification to determine performance characteristics, thereby enabling selection. Briefly, equations involved are as follows:

$$(1) \qquad \frac{h_a A_s}{.243\ W_a} = \log_e \frac{t_D - t_D''}{t_a - t_a''} = \log_e R$$

$$(2) \qquad T_s = \frac{R t_a - t_D}{R - 1}$$

$$(3) \qquad H_T = \frac{h_r}{R_s} A_s (T_s' - T_r)$$

where

A_s = Air side surface in square feet.
h_a = Air side heat transfer coefficient.
h_r = Refrigerant side heat transfer coefficient.
H_T = Total coil load in Btu per hour.
$R = \frac{t_D - t_D''}{t_B - t_B''}$
R_s = Ratio of air side surface to refrigerant side surface.
t_B'' = Dew point temperature of air leaving coil.
t_B = Dry bulb temperature of air leaving coil.
t_D'' = Dew point temperature of air entering coil.
t_D = Dry bulb temperature of air entering coil.
T_r = Refrigerant saturation temperature corresponding to coil suction pressure.
T_s = Surface temperature of coil.
W_a = Pounds per hour of air-vapor mixture flowing across coil.

The following factors were obtained for the specification coils:

	Plate Fin Coil	Integral Fin Coil
h_a	$1.08\ (\rho\ V)^{.65}$	$1.41\ (\rho\ V)^{.65}$
h_r	250-350	300-400
A_s	A_f (18.4N-7)	1.3nL
A_r	0.145nL	0.145nL

where

A_f = Net face area of finned section of coil in square ft.
A_r = Refrigerant side surface area in square ft.
L = Length of each tube in feet.
n = Number of tubes in coil.
N = Number of tube rows in coil depth.
$(\rho\ V)$ = Air density in pounds per cubic feet times air velocity in feet per minute.

It was necessary to simplify performance data before they could be issued to the field; and fortunately repeated use of the formulae eventually led to the development of a slide rule type selector. The use of the selector requires slight familiarity with the psychrometric chart and does not require application of any of the above formulae. Mathematical calisthenics are reduced to simple arithmetic.

By following a similar routine, Case then developed a specification for a gravity coil. Gravity coils must be used in magazines because of the stringent regulations regarding electrical equipment and may also be used elsewhere where heat loads do not warrant the use of coils requiring forced circulation. Five (5) coils from various manufacturers were tested under simulated shipboard conditions. Analysis of these tests resulted in a specification for a two row coil of staggered tubes, .016″ plate copper fins uniformly spaced 3 fins per inch. Details were furnished for the drain pan design.

General type plans for 6 sizes of duct type Freon-12 coils, 6 sizes of unit coolers and 3 sizes of gravity coils were prepared. Complete performance and dimensional details were furnished all Naval Design personnel and contracts were let to establish stocks of each item.

Numerous tests were also conducted at Case on fabricated Navy Standard Cooling equipment to check performance of test coils against that of production equipment. Manufacturers were advised to change many of their construction details such as, distributor port size, blade pitch of unit cooler fans, size and location of baffles to improve air distribution of unit coolers and to observe strict compliance with fin spacing specifications.

By early 1945, it was decided to develop a chilled water cooling coil against the possibility of using chilled water as the refrigerant instead of Freon-12. Earlier experiments had established an optimum air side surface so that the later series of tests were concentrated on the chilled water circuiting details, determination of the necessity for and the optimum design of turbulence devices, methods of capacity determination and the development of a unit cooler which would be rectangular rather than square to minimize use of available head room. Two coils identical except for circuiting were built and tested. Test results indicated that high coefficients of heat transfer could be maintained at low water velocities by the use of spiral springs. This made it possible to retain the simple circuiting of four tubes per circuit, with all return bends horizontal. While the same high heat transfer could also be obtained by the use of high water velocities, the low water velocities in combination with turbuspirals reduced the overall weight of the system without appreciably increasing the pumping head. Considerable work was involved in estab-

lishing a method of determining heat transfer capacities of the chilled water coils for selection purposes in the field and again a slide rule type selector was developed.

Development of a rectangular unit cooler will be completed by the matching of the best currently available fans to the units. Tests comparing blow through versus draw through types, various depths of plenum chambers and different types of straightening vanes resulted in acceptance of the blow through type which incorporates a transition piece made up of a plenum chamber, fixed straightening vanes and an integral easily removed filter. Six sizes of chilled water coils, all eight rows deep, have been established. Four of these sizes with appropriate transition pieces and fans will make up the chilled water unit cooler family. It is expected that ultimately the same fans and transitions will be incorporated into a new Freon-12 unit cooler family. All chilled water coils are completely insulated with removable section of drain pan facilitating cleaning.

In order to obtain complete control of the air conditioning process with fixed air distribution, it is necessary to adjust for moisture changes as well as for dry bulb changes. However, since accurate simultaneous control of wet bulb and dry bulb requires complex control equipment or facilities for dew point control and reheat, it was decided to investigate the satisfactoriness of simple dry bulb control. Both on-off control and proportional refrigerant flow mechanisms were tested and it was demonstrated that while both methods could be made to hold satisfactory room conditions, sensitivity of currently available commercial controllers makes on-off control more desirable, at least for the present. In on-off control, controller action is cyclic, in other words, when the dry bulb thermostat is satisfied, refrigerant flow to the coil is stopped and the relative humidity in the compartment rises. Both dry bulb and wet bulb temperatures will rise until the thermostat again permits the flow of refrigerant to the coil. Of course, so long as refrigerant is permitted to flow through the coils, there is water removal. It has been demonstrated that acceptable cycling of relative humidity will result only if the thermostat is sufficiently sensitive to maintain the cycling dry bulb temperature change in the room to an amplitude of 2-3° F. The thermostat selected will be required to maintain this differential.

PHYSIOLOGICAL RESEARCH.

Supplementing the evaluation of service experience as hostilities progressed, the Bureau of Medicine and Surgery undertook a series of physiological studies to determine the need for shipboard air cooling and the minimum extent air conditioning need be applied to maintain personnel efficiencies. Two series of tests were conducted at the Naval Medical Research Institute, Bethesda, Md.; one series aboard the U.S.S. *Franklin*, a CV9 class aircraft carrier; two series on the most modern of the hospital ships; and one series on the U.S.S. *Washington*. These tests clearly established the following basic points:—

(a) That men forced to work in a hot space will perform tasks significantly better if they are provided with cool rather than hot living and sleeping quarters.

(b) That design should be based on maintaining a maximum of 78° effective temperature in compartments. 78° ET will maintain quarters just below the sweating threshold when personnel therein are at rest.

(c) That heat rash which invariably develops in ventilated compartments while cruising in tropical areas can be prevented by daily part-time cooling at 78° ET.

The first series of tests at the Research Institute was carried out on two groups of 10 volunteers. Each test was conducted for ten days without interruption. Both groups worked for 7 hours each day in a simulated "work space" maintained at 108° F. dry bulb and 83° F. wet bulb or 90° ET. These conditions are somewhat hotter than ventilated shops but are comparable to radio transmitter rooms, spaces such as catapult or crane machinery rooms and areas immediately adjacent to blast cooled watch stations in engine and boiler rooms. The hot group spent the remaining 17 hours of the day in "living quarters" maintained at 95° F. dry bulb and 83° F. wet bulb or 87° ET which represents good ventilation conditions for crews berthing compartments. The cool group spent the remaining 17 hours in living quarters maintained at 85° F. dry bulb and 71° F. wet bulb or 78° ET. Work for each man in both groups consisted of walking at the rate of 4½ miles per hour for 5 minutes, 8 times during the daily work periods. Studies included the evaluation of 32 physiological and psychological functions. Examination of

such diversified functions as sensori-motor performance (hand dynamometer endurance, photoelectric aiming, pursuit rotor, discrimination reaction time, etc.), various aspects of vision, and the performance of physical work in the heat consistently demonstrated the superiority of the cool group. Clinical observation indicated that the physical condition of the men resting in cool quarters was better than that of men resting in hot quarters. The men in the cool quarters slept better and were negligibly bothered by heat rash which was almost universal in the group resting in hot quarters.

A second series of tests was conducted at the Research Institute to more carefully investigate the effects of acclimatization. This series also involved closer control of diet, improved techniques of measurements and somewhat milder ambient conditions. Two groups of 6 men each, all 18 years old, lived in a temperature environment of 85° ET (90° F. dry bulb and 83° F. wet bulb) for 9 hours daily. Both groups performed treadmill work tests at 90° ET (108° F. dry bulb, 83° F. wet bulb) for 3 hours daily. Work consisted of walking at the rate of 4½ miles per hour for 5 minutes, twice during each work period. The hot group slept in an 85° ET ambient while the cool group slept in a 75° ET ambient (80° F. dry bulb; 70° F. wet bulb) for the remaining 12 hours. The men were exposed to these conditions for 30 days, preceded by an 8 day control period and followed by a 6 day recovery period.

The scope of the physiological and psychological tests was similar to the first series of tests. The hot group again exhibited extensive heat rash which continued throughout the 30 day period and disappeared at the end of the 6 day recovery period. Heat rash in the cool group was negligible. While sensorimeter functions showed little difference between the two groups, evidence indicated that the higher physiological cost, even to 18 year old men in good physical condition, of living in the hotter ambient conditions might lead to deterioration with longer exposure. Subjectively, it was noted that the men in the cool group moved around in their quarters, washed their clothes, wrote letters and exhibited alertness that was lacking in the hot group.

Measurements of pulse rate, blood pressure and oral temperature were made on 20 men aboard the *Franklin* during her

shakedown cruise and continuous recording of berthing compartment and weather dry bulb and humidities were taken. Weather air conditions were fairly constant for about 20 days at 77-79° ET while the berthing compartment varied from 81-85° ET. Exposure to the heat was comparatively short and compartment dry bulb temperatures of 85-88° F. were mild for tropical condition. Compartment ventilation was excellent, as evidenced by the 7.8° F. mean dry bulb temperature rise above weather air. Nevertheless, almost all of the crew complained of broken sleep due to heat, of feeling tired and "dopey" on awakening. There was little if any heat rash probably due to intermittent cooling at 78° ET topside in the weather air. The crew performed delegated tasks efficiently but with greater individual effort and reduced initiative than is observed in temperate environments.

The series of tests on the early fully air conditioned hospital ships and on the *Washington* were accomplished under ambient conditions quite similar to the later test at Bethesda and resulted in the same general conclusions. On at least one occasion, air conditioning of the hospital ships involved unnecessary red tape. The customs official inspecting one of the ships in Panama on a hot sultry day did a very careful and complete job. The ship was simply too comfortable for him to go ashore. On the same ship, there was a notable absence of requests for shore leave. The crew preferred to stay aboard too. The shipboard tests however did emphasize the following:

(a) Cross connection of heat exchange machinery to avoid complete breakdown of any given system is necessary. In the event of complete failure of an air conditioning system the remaining cooling effect would be that produced by replenishment air at the rate of 5 to 15 cfm per person as compared to 30-60 cfm per person in compartments fitted for cooling by ventilation.

(b) Excessively chilled atmospheres must be avoided.

One of the principal factors in the design of air condition systems for shipboard application is the determination of the absolute minimum quantity of replenishment air necessary to keep odor levels within the various compartments at a reasonably acceptable level. This information was obtained by the Naval Research Institute in what is probably the most extenisve test of its kind conducted to date. Prior to these tests outstanding air conditioning authorities and physiologists offered requirements

ranging from 10 to 20 cubic feet per minute per man of replenishment air for shipboard installations. In compartments where personnel account for a large proportion of the total load, the air conditioning equipment necessary to cool the replenishment air from the weather condition to the desired room ambient is appreciable. Lack of specific information, and weight and space limitations aboard ship, ultimately developed to the point where engineers were forced arbitrarily to decrease replenishment air requirements. For example, control spaces on combatant vessels usually are provided with 15 cfm per man of replenishment air while berthing areas on the hospital ships are provided with as little as 7 cfm per man. Some of these reductions were made not without consternation and there was a general reluctance to proceed on a large scale without definite facts.

To establish these facts, 22 separate replenishment air quantity tests were conducted each lasting from 2 to 17 days. Considerable care was taken to duplicate shipboard conditions. The so called "battleship" compartment was fitted to accommodate 68 bunks and lockers (95 cu. ft. per bunk). 45 men were always in the compartment, all soiled clothing was stowed in standard lockers within the compartment. The compartment was swept twice a day and swabbed down once a day. All bed linens, mattresses and mattress covers were the same as those used aboard ship. Each man exercised for 4 hours a day. Exercise was sufficient to work up a sweat, after which a shower was permitted. General toilet was permitted in the morning and after the evening meal. One-half the clothes were laundered each week. Smoking was generally permitted.

The compartment was fitted with an air lock and replenishment air carefully controlled and measured. Quantities of replenishment air were varied from 35 cfm per bunk to as little as ⅓ cfm per bunk.

Conclusions reached as a result of the replenishment air tests are as follows:

(a) There are three basic odor impressions. The first, an immediate impression upon entering the compartment; the second, the impression recorded after one minute of exposure and a third, the impression incident to prolonged occupancy. The immediate impression of odor is materially higher than the one minute impression and this impression gradually disappears.

(b) 3 cfm of replenishment air per bunk at 78° ET will result in acceptable primary exposure levels provided the cooling coil is condensing water. The one minute exposure level at this rate of replenishment air is barely above the odor perception threshold. Further, the quality of air at 3 cfm per bunk and 78° ET with wet coils is comparable to that attained with 5 cfm per bunk when the coils are dry or with 35 cfm per bunk of replenishment air at 81° ET utilizing ventilation without cooling.

(c) The replenishment air rate can be further reduced to ⅔ cfm per bunk by the utilization of activated carbon.

(d) Neither a well advertised liquid chemical deodorant nor ozone reduced odor levels, although a 10 minute exposure to either exerted an anesthetic effect upon odor perception ability.

(e) Carbon dioxide, carbon monoxide, bacteria and dust did not reach objectionable or even approach noxious levels at as little as ⅔ cfm of replenishment air per bunk. The particular importance of this item is that it is an additional demonstration that the possibility of carbon dioxide buildup or oxygen depletion is remote for most of the occupied compartments of surface vessels. Shipboard tests have repeatedly confirmed the finding that the opening and closing of doors for normal access, feeding or sanitary purposes even when restrained to the absolute minimum during general quarters provides sufficient leakage to prevent CO_2 buildup and O_2 depletion. Soda lime trays and oxygen tanks had been removed and will not be replaced in future designs utilizing the reduced replenishment air rates.

Experimental Installations.

Wartime experience, especially in the South Pacific, with ventilated berthing compartments ultimately culminated in authorization by the Chief of Naval Operations to incorporate experimental air cooling equipment in two cruisers now under construction. Vice Admiral E. L. Cochrane, wartime Chief of the Bureau of Ships, has emphasized that the Navy, "Is not trying to mollycoddle its personnel. Admittedly, the ships will be more comfortable in tropical climates, but that consideration is not paramount. It is to improve the battle effectiveness that this is being done, further, the fighting efficiency of our Fleet is lowered in proportion to the human strain and exhaustion caused by exposing personnel to long periods of abnormal heat. There

can be no question but that the crew in an air-conditioned warship will have a distinct and perhaps a decisive advantage in action with an enemy fatigued and of lower alertness from living in a ship that is not air-conditioned. These installations are experimental only insofar as the equipment itself is concerned. The rigors of tropical warfare pose a tactical problem that can be solved only by air conditioning all of our major combatant ships in the future. For the present we are seeking to determine what type of equipment is most efficient for the purpose."

Under this experimental program, both steam jet refrigeration equipment and high speed Freon-12 compressors will be installed. The U.S.S. *Salem* when completed in late 1947 at the Bethlehem Steel Co. Shipyard, Quincy, Mass., will be provided with three steam jet plants each of about 65 tons capacity serving hotel spaces and three Freon-12 compressor condensers of conventional design with Freon-12 to water heat exchange apparatus to permit the use of chilled water cooling coils throughout the ship, and in effect, provide steam jet plant standby capacity for the vital plants. Three steam jet plants designed and built by different manufacturers will be tested at the Engineering Experiment Station, Annapolis, Md., and if the tests warrant it, these three plants will be installed on the *Salem* to obtain varied maintenance and operational experience. The U.S.S. *Newport News*, now under construction at the Newport News Shipbuilding and Dry Dock Company at Newport News, Virginia, and also scheduled for completion in late 1947, will have equipment utilizing high speed Freon compressors with direct expansion Freon-12 cooling coils. On the U.S.S. *Newport News*, refrigeration compressors will also be the high speed type not only to reduce the size and weight of the equipment but also to effect interchangeability of spare parts with those of the air cooling equipment. The air conditioning design requirements for these two cruisers are such that the compartments to be mechanically cooled, the quantity of replenishment air, and the air conditioning tonnage in each compartment are essentially identical to provide a direction comparison of the merits of the two types of installations.

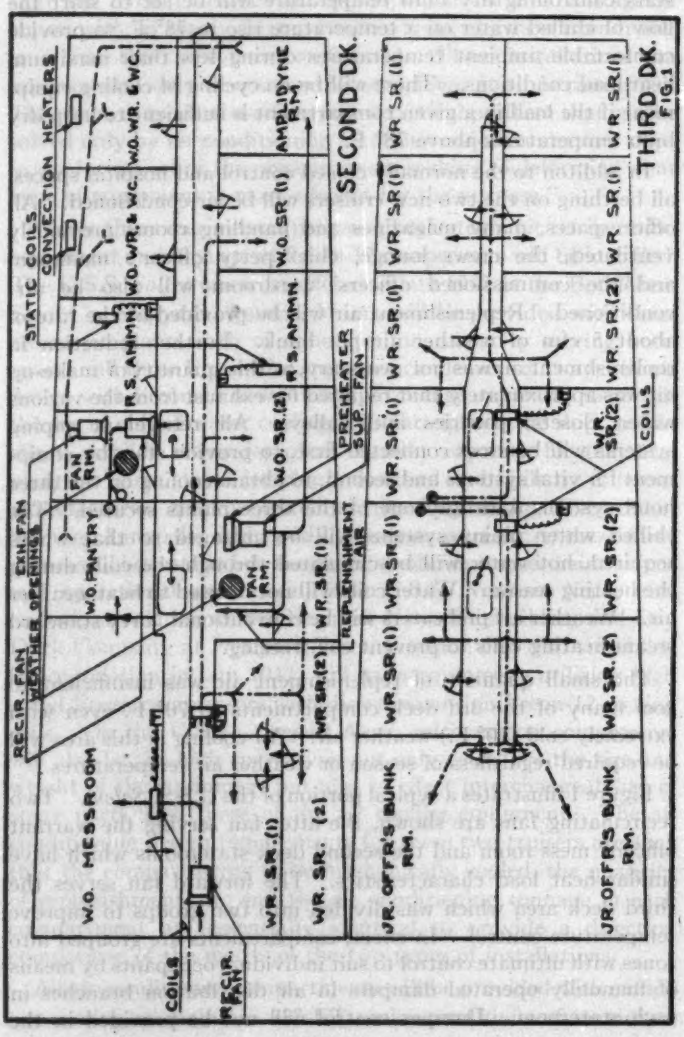
All air-conditioned compartments will be provided with sufficient cooling to maintain 78° ET under conditions of maximum load. This is equivalent to 85° F. dry bulb at about 50 per cent relative humidity and while outside of the comfort range, will

prevent perspiration during light activity or at rest. Thermostats controlling dry bulb temperature will be set to start the flow of chilled water on a temperature rise to 78° F. to provide comfortable ambient temperatures during less than maximum heat load conditions. There will be no cycling of cooling equipment if the load in a given compartment is sufficient to keep dry bulb temperatures above 78° F.

In additon to the normally cooled control and hospital spaces, all berthing on the two new cruisers will be air-conditioned. All office spaces, many magazines and handling rooms previously ventilated, the crews lounge, chief petty officer's messroom and the commissioned officers' wardroom will also be air-conditioned. Replenishment air will be provided at the rate of about 5 cfm of weather air per bunk. Further reduction in replenishment air was not necessary, as this quantity of make-up air was approximately that required for exhaust from the various water closets, pantries and galleys. All refrigerant piping systems will be cross connected first, to provide standby equipment for vital systems and second, to obtain cooling on the three hotel systems with any one of the three plants secured. The chilled water piping systems will be arranged so that where required, hot water will be circulated through the coils during the heating season. Water coils will not be used to heat weather air. Weather air preheaters will be conventional Navy standard steam heating coils to prevent coil freezing.

The small quantity of replenishment air was insufficient to cool many of the 3rd deck compartments to 70° F. even with extremely cold (10° F.) weather air. Air cooling in this area will be required regardless of season or weather air temperatures.

Figure 1 illustrates a typical portion of the U.S.S. *Salem*. Two recirculating fans are shown, the after fan serving the warrant officers' mess room and the second deck staterooms which have similar heat load characteristics. The forward fan serves the third deck area which was divided into two groups to improve temperature control. In effect, compartments are grouped into zones with ultimate control to suit individual occupants by means of manually operated dampers in air distribution branches in each stateroom. Damper control will not be provided in the messroom since this compartment utilizes the full capacity of the coil supplying it and temperature control is by its own thermo-


SECOND DK.
THIRD DK.
STATIC COILS
CONVECTION HEATERS
W.O. WR. & WC.
W.O. WR. & WC.
AMPLIDYNE RM.
W.O. SR (1)
R.S. AMMN.
R.S. AMMN.
W.O. SR.(2)
PREHEATER SUP. FAN
FAN RM.
W.O. PANTRY
EXH. FAN
WEATHER OPENINGS
RECIR. FAN
FAN RM.
W.O. SR.(1)
REPLENISHMENT AIR
W.O. MESSROOM
WR. SR.(2)
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stat. The single supply fan in this area provides replenishment air for all coils and also ventilates the amplidyne room which was not mechanically cooled. The single exhaust fan removes a portion of the supply to the mess room via the pantry, exhausts all of the replenishment air via the two washrooms and water closets and also exhausts the ventilated amplidyne room. The preheater will be a steam heater, reheat will be accomplished by circulating warm water in the chilled water piping system when necessary, and convection type steam heaters will be installed in the washrooms and water closets. Also illustrated are the static type gravity coils to be installed in the ready service ammunition spaces.

In addition to the heavy cruisers *Salem* and *Newport News*, there is a third sister ship and a class of light cruisers under construction which while not completely air-conditioned will have air conditioning in 3rd deck living compartments in way of machinery spaces. These vessels will be fitted with conventional Freon-12 direct expansion equipment. These different types of machinery and the wide variation of extent of applied cooling will furnish the foundation for decisions from a design, maintenance, operational and fleet viewpoint as to the ultimate extent and type of equipment to be used in mechanical cooling aboard warships.

The *Salem* and *Newport News*, while referred to as totally air-conditioned ships, will not have mechanically cooled air in machinery spaces and many shops and messing compartments will remain ventilated. It is within the bounds of possibility, however, that future considerations will necessitate provisions for recirculation of all the air in the ship for extended periods. These considerations may ultimately necessitate further extension of mechanical cooling even to the inclusion of machinery spaces.

EVALUATION OF HIGH TEMPERATURE GREASES[1].

By Robert C. Adams[2] and Sidney M. Collegeman[3].

The primary purpose of the testing of lubricants, of which high-temperature greases are only a single class, is the determination of their suitability for specific applications. Of almost equal importance, although this secondary nature is not always clearly recognized, is obtaining the evaluation quickly, economically and reproducibly. During the press of war the urge for speed was so overwhelming that a greater reliance than desirable had to be placed on judgment and hypothesis instead of upon data. It should be possible, however, to base decisions upon the results of tests whose reliability and significance are subject to accurate appraisal. This paper presents a history of the development by the Navy of methods for the evaluation of high-temperature greases and outlines the program for future study to be undertaken by the Experiment Station.

Evaluation should precede specification, but ten years ago the only accepted methods of test were those of the ASTM, which even today are inadequate for identification, much less for evaluation. However, specifications were constructed on this rickety base, although a statement of limits for manufacturing formulas and such properties of the finished product as penetration and dropping point should more properly be termed a *prescription*. The evident lack of evaluation in the prescription or composition type of specification invites the unscrupulous manufacturer to offer the cheapest product which will meet the more discriminating requirements and it circumscribes the efforts of the competent and progressive manufacturer who is continuously improving the quality of his product. Military demands for reliable greases suitable for the extreme conditions of continuous duty at high rates required of fighting vehicles, aircraft and ships evoked a new concept in grease specifications.

(1) This paper was presented before the Annual Meeting of the National Lubricating Grease Institute, October 1946. The opinions expressed herein are those of the authors and are not necessarily official opinions of the Engineering Experiment Station or the Navy Department.

(2) Assistant Superintendent, Chemical Laboratory, U. S. Naval Engineering Experiment Station, Annapolis, Md.

(3) Leader, Grease Unit, Petroleum Section, U. S. Naval Engineering Experiment Station, Annapolis, Md.

The genesis of performance specifications for greases was discussed in 1941 (4). The fallibility of composition specifications had been recognized earlier and the need for better lubricants had been partially satisfied by accepting the recommendations of equipment manufacturers. These recommendations were based upon trials which the manufacturers had made of various greases for lubricating their products. The armed services thus undertook to maintain stocks of many proprietary greases for the variety of devices which they use. This policy developed an intolerably complex supply problem. It is not reasonable that two generators, or reduction gears, or plug valves, of similar service, rating, speed and temperature should require different greases simply because they came from different factories.

In an effort to clear this fog it was reasoned that if the torque necessary to restrain the outer race of a lubricated bearing on a rotating shaft could be measured accurately, the resulting graphs of torque vs. speed, torque vs. time and torque vs. temperature could be correlated with the service record of greases which had been used successfully to provide a rapid method of evaluation. The device constructed to carry out this idea was the Navy Grease Machine illustrated in Figure 1. Experience showed that the only way to obtain reproducible results with the machine was first to insure uniform packing of grease and then to operate at constant speed and temperature. The parts of the bearing packer also are shown in Figure 1. Complete delineation of the four dimensions of torque, time, temperature and speed was intolerably time-consuming so that specimen conditions of temperature and speed had to be selected. Two sets of conditions were employed: 600 Rpm. at 100° F. and 3000 Rpm. at 400° F. (later 350° F.). It was also found that deterioration of the grease during service at high temperature could not be measured on the small amount of residual lubricant in the bearing of the Navy Grease Machine. The Navy Grease Beater, shown in Figure 2, was built to provide accelerated oxidation of grease somewhat similar to deterioration in service. In the Beater approximately half a pouud of grease could be recirculated through a rapidly rotating bearing while being maintained at an elevated temperature (400 or 350° F.). After oxidation in the beater, the beater grease was retested in the Navy Grease

(4) Proceedings American Society for Testing Materials, Volume 41, Pages 1095-1104 (1941).

Machine to determine the change in its properties as reflected by the torque-time curves.

Demands for new guidance in purchasing were so insistent that it was not possible to confirm the validity of NGM criteria by correlation with service tests. Instead, the first specification was prepared by providing empirical limits above and below the torque-time curves traced by what appeared to be the best of several greases tested. Reports of unsatisfactory high-temperature grease from Naval aircraft decreased gratifyingly and the improvement was prematurely attributed to the new specification.

The grease was better than the Navy had obtained previously but only one manufacturer, that of the prototype, qualified under the specification. Because of the need for speed in wartime, coupled with the fact that only one product appeared in the bulletin of approved materials, inspection tests of early deliveries were omitted. The Navy Grease Machine and Beater were busily engaged in qualification tests for other manufacturers with consistently unsatisfactory results. Eventually an inspection test was made on a delivery of the approved material and it failed! Obviously the tests of the specification did not measure the significant properties of the grease.

No other equipment for testing high-temperature aircraft greases was available so that the urgently needed revision of the specification had to be based on the Navy Grease Machine and Beater. This temporary specification was an outright hypothecation. It was assumed that the most desirable grease is the one which shows the least change in torque with time or with exposure to high-temperature conditions. Specification requirements consisted of a series of maximum permissible ratios between initial torque and final torque on the fresh grease and beaten grease and between the torque measured on the beaten grease and the corresponding value from the NGM run on the fresh grease. Although these criteria are reasonable, we were not too sanguine about the service suitability of products satisfying them. There had been enough of rationalization in the development and some demonstration and data on service performance were needed.

The first step was the design and construction of a new apparatus, the High Temperature Grease Apparatus of Specification ANG5a, shown in Figure 3, in which service operating conditions could be simulated. Cooperation of the CLLG Group of

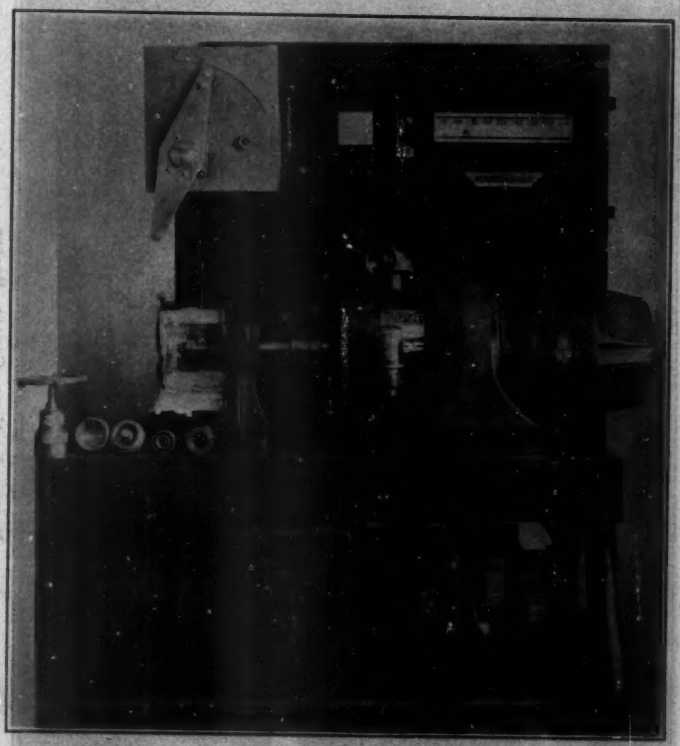


FIGURE 1.—NAVY GREASE MACHINE AND BEARING PACKER.

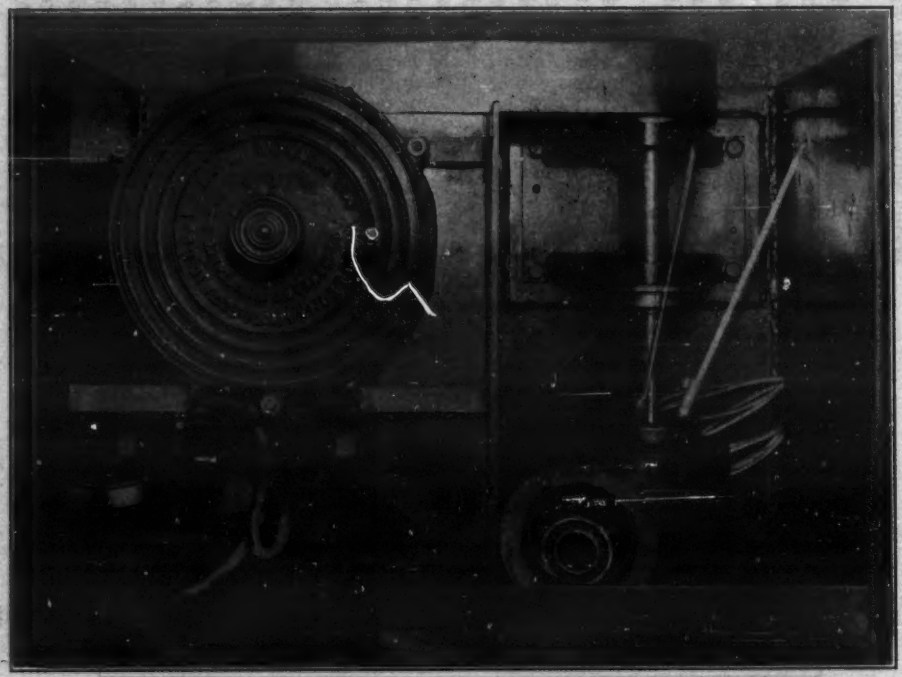


FIGURE 2.—NAVY GREASE BEATER.

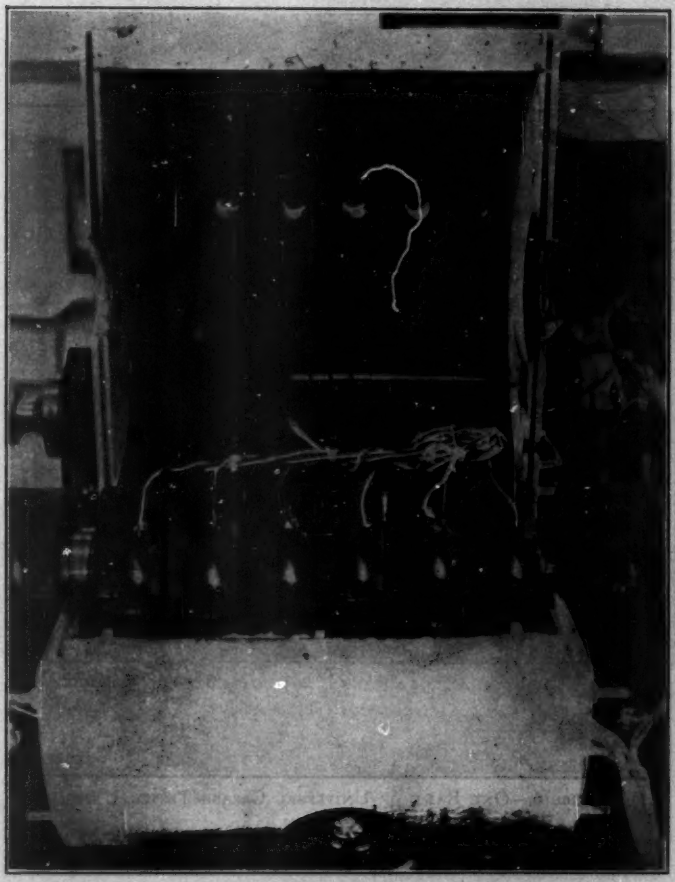


FIGURE 4.—MULTIPLE BEARING GREASE TESTING UNIT (CENTIPEDE). ONE OF SIX UNITS.

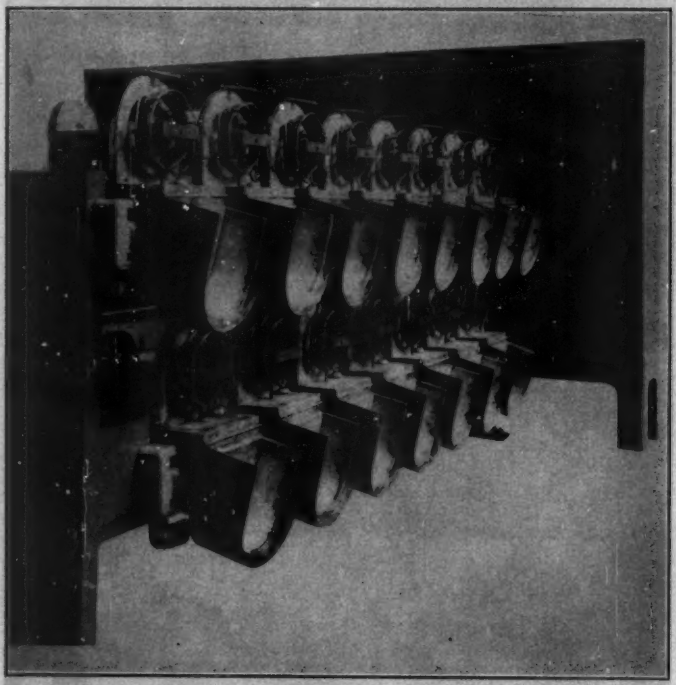


FIGURE 6.—ONE BANK OF UNIVERSAL GREASE TESTER UNITS, OVENS OPEN.

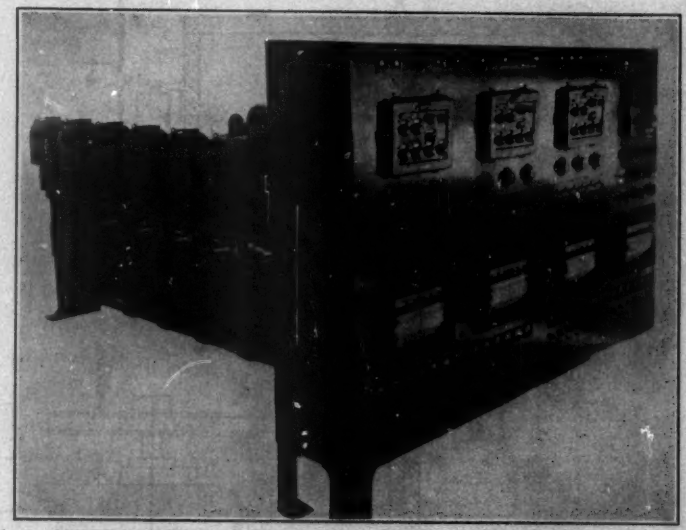


FIGURE 7.—END OF CONTROL PANEL AND BANK OF UNITS, OVENS CLOSED.

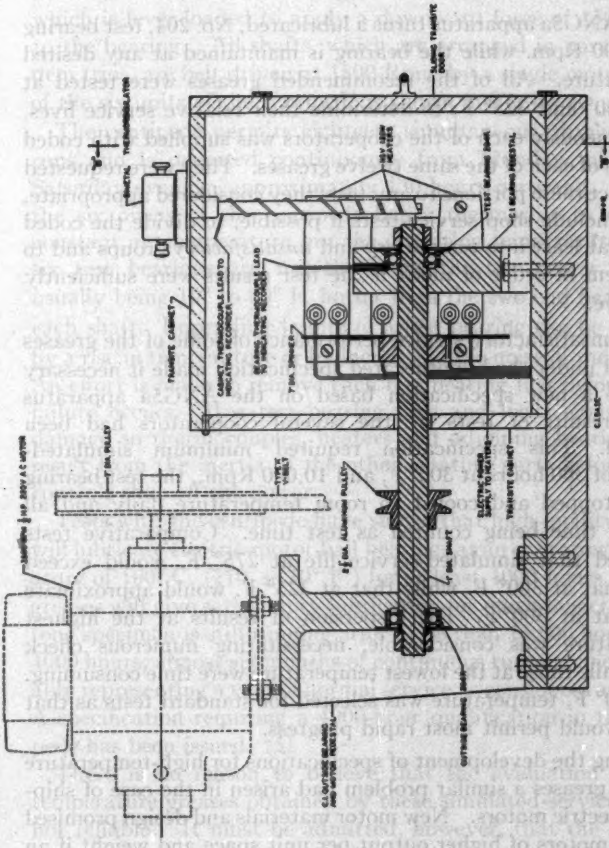


FIGURE 3.—HIGH TEMPERATURE GREASE APPARATUS, SPECIFICATION ANG 5A.

the Coordinating Research Council, grease makers and manufacturers of aircraft-engine accessories was obtained both for supply of samples of recommended greases and for evaluation of these greases by both laboratory tests and shop trial in actual equipment.

The ANG5a apparatus turns a lubricated, No. 204, test bearing at 10,000 Rpm. while the bearing is maintained at any desired temperature. All of the recommended greases were tested at 275°, 300° and 325° F. to determine their relative service lives. Simultaneously each of the cooperators was supplied with coded portions of each of the same twelve greases. They were requested to subject these portions to any tests they considered appropriate, but to include shop service-tests if possible, to divide the coded greases at least into *satisfactory* and *unsatisfactory* groups and to rate them in order of merit if the test results were sufficiently definitive.

The unsatisfactory service performance of some of the greases supplied under the hypothecated specification made it necessary to issue a new specification based on the ANG5a apparatus before results of tests by the several cooperators had been received. This specification required minimum simulated-service of 300 hours at 300° F. and 10,000 Rpm., the test bearing being stopped and cooled to room temperature daily and all running time being counted as test time. Comparative tests indicated that simulated-service life at 275° F. would exceed twice that at 300° F. while that at 325° F. would approximate half that at 300° F. The dispersion of results at the highest temperature was considerable, necessitating numerous check runs, while those at the lowest temperature were time consuming. The 300° F. temperature was selected for standard tests as that which would permit most rapid progress.

During the development of specifications for high-temperature aircraft greases a similar problem had arisen in the case of shipboard electric motors. New motor materials and design promised electric motors of higher output per unit space and weight if an increase in bearing temperature to 100° C. or more could be tolerated. It was desirable to take advantage of this possibility but, "once burned twice shy", the determination should be based upon simulated-service tests and not upon results of accelerated tests not verified as representing service conditions.

A mulitple-bearing apparatus, familiarly called the *centipede*, was built for the comparison and appraisal of electric-motor-bearing greases. The apparatus consists of six rotating shafts each mounted in a separate oven and carrying six, No. 212 bearings. Each of the bearings is enclosed in a cast-iron housing which is lever-loaded to apply a downward force of 150 pounds to the bearing. All shafts, which are arranged in parallel, tandem trios, are belt driven at 1800 Rpm. by a single motor. One of the six units of the centipede is shown in Figure 4.

The centipede permits eighteen, simultaneous, duplicate test runs and is operated continuously from Monday morning to Saturday evening, approximately 130 hours per week. Each of the six ovens of the centipede has a thermostat to maintain constant air-temperature but the operating temperatures of the six test bearings are not the same, the two center bearings usually being 10° to 20° F. hotter than the two end bearings on each shaft. Impending failure of a test bearing can be detected by a rise in temperature or by increase in the noise of the bearing. An effort is made to remove each test bearing just before actual failure occurs. If a test bearing fails and locks, considerable damage to thermocouples, heaters and adjoining bearings may result from the inertia of the other rotating parts before safety controls shut off the power.

Tests with the centipede have shown that high quality greases will lubricate electric-motor ball bearings at an operating temperature of 100° C. (210 ± 10° F.) for at least 4000 hours. Many greases will give satisfactory lubrication for much longer periods, (one specimen is still running after more than 15,000 hours), but 4000 hours, almost six months of continuous running and reasonably representing a year of normal service, is considered adequate. A specification requiring a 4000-hour qualification in the centipede has been issued. (5)

There is no reason to believe that the evaluation of high-temperature greases obtained by these simulated-service tests is not reliable. It must be admitted, however, that the tests are time-consuming, particularly the centipede tests which require about eight months. The most pressing need, therefore, is for a means of accelerating the testing so that results can be obtained in a shorter time. Ideally, it should be possible to apply the test

(5) Navy Department Specification 14-L-3c; Lubricant, Ball and Roller Bearing; 15 Jan. 1946.

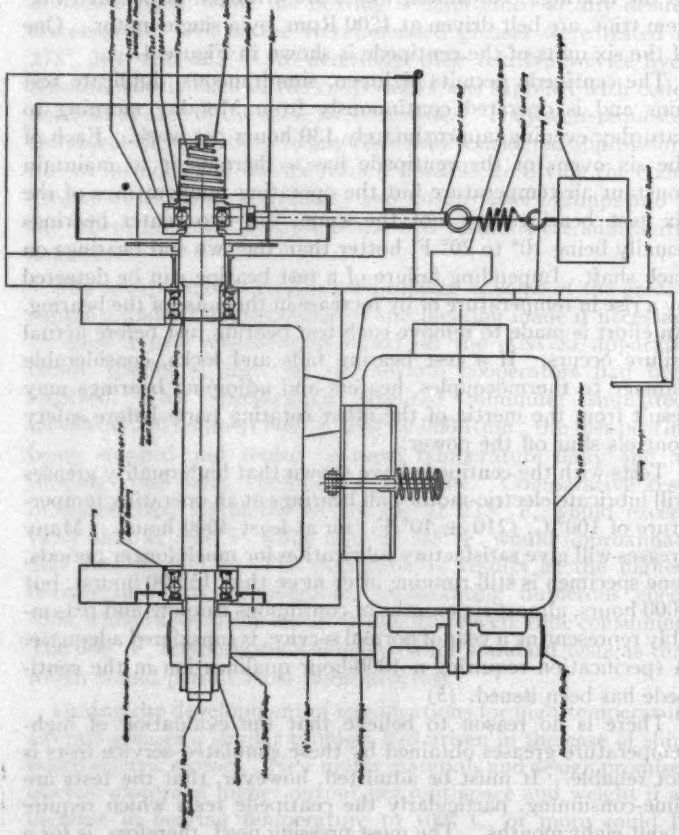


FIGURE 5.—UNIVERSAL GREASE TESTER UNIT. SECTION ON CENTER LINE.

for inspection of each delivery lot, which would require reduction of testing time to not more than three days. Such a manifold acceleration of testing conditions and reduction in testing time must be achieved by many successive steps, each corroborated by correlation with the results of the preceding step, if the present reliable evaluation is not to be imperiled. Furthermore, each step from simulated-service to highest acceleration must be conducted in the same testing machine if past history of a new design and a new testing device for each new problem are to be avoided.

In an effort to provide this Universal Grease Tester the Station has designed the unit illustrated in Figure 5. This unit recognizes and provides control for four variables: temperature, speed, bearing size and bearing load. Permissible temperatures with the present installation range from room up to 500° F.; speed, up to 20,000 Rpm.; bearing size, up to 60 mm bore; and bearing load, up to 50 pounds. Temperature and load ranges could be extended readily by minor modification of accessories without any essential change in the testing unit.

Thirty Universal Grease Tester units have been installed in a compact assembly wherein one operator with semi-automatic controls can achieve approximately 600 bearing-test hours per day. This installation is illustrated in Figures 6 and 7. The multiple installation is necessary in order to obtain in reasonable time sufficient replicate runs for appraisal of both the magnitude and the variability of the results.

The program for study of high-temperature aircraft greases illustrates the projected use of the Universal Grease Testers for development of rapid, reliable methods for performance evaluation of anti-friction bearing lubricants. The greases currently approved under Specifications ANG5a and 14L7 each have been subjected to quadruplicate tests under the conditions of those specifications: 300° F., 10,000 Rpm., No. 204 bearings. A modified control-chart analysis (6) has been applied to the results. This is an admitted misapplication of control charts, which were developed for detection of significant variation in successive lots of presumably identical material, but the objective and manner of their use it is hoped are not an undue violation of mathematical principles.

The mean, $\overline{X}$, and standard deviation, σ, of each set of four

(6) Supplement B, ASTM Manual, in Presentation of Data, 1945 Reprint.

tests are calculated and the results plotted on a control chart. Figure 8 illustrates this process with the first thirteen greases tested. The central line in the upper array is the grand average, $\overline{\overline{X}}$, of the several individual means and that in the lower array is the mean, $\overline{\sigma}$, of the several standard deviations. Control limits calculated from $\overline{\overline{X}}$ and $\overline{\sigma}$ (6) are shown as broken lines.

Present interpretation of the results shown in Figure 8 is the following:

1. There is no indication of a significant difference in acceptability or expected service performance among the greases whose $\overline{X}$ values lie within the control limits.
2. Greases whose $\overline{X}$ values fall below 275 hours, the lower control limit, are significantly poorer than the average of all greases tested. (In this case none of the greases fell below this figure).
3. Greases whose $\overline{X}$ values are greater than 760 hours, the upper control limit, are significantly better than the average of all greases tested (Greases B and G).
4. Any results on a grease whose standard deviation exceeds 300, upper control limit for σ, are questionable. Such an excessive dispersion in results had it occurred, would suggest that operational errors in selecting, cleaning, packing or mounting test bearings had occurred. Such a set of results should be discarded and another quadruplicate test made.

When such a set of data as shown in Figure 8 is completed, one variable can be intensified and a similar set of data obtained under the accelerated conditions. Only one variable at a time should be changed, to avoid confusion, and the change should be reasonable in amount. In the case of high-temperature aircraft grease, which has served as the example, the first change will be to increase the operating temperature from 300° to 325° F.

When two sets of data differing in only one variable are available, a sound decision as to the interchangeability of the two sets of testing conditions can be made. If the same greases are significantly poorer, and the same greases significantly better, than the average and if the dispersion of results is not greatly different, then there is no reason why the more severe conditions cannot be used as satisfactorily for specification requirements with a reduction in testing time. Review of the data also will indicate whether further acceleration by increase in temperature is promis-

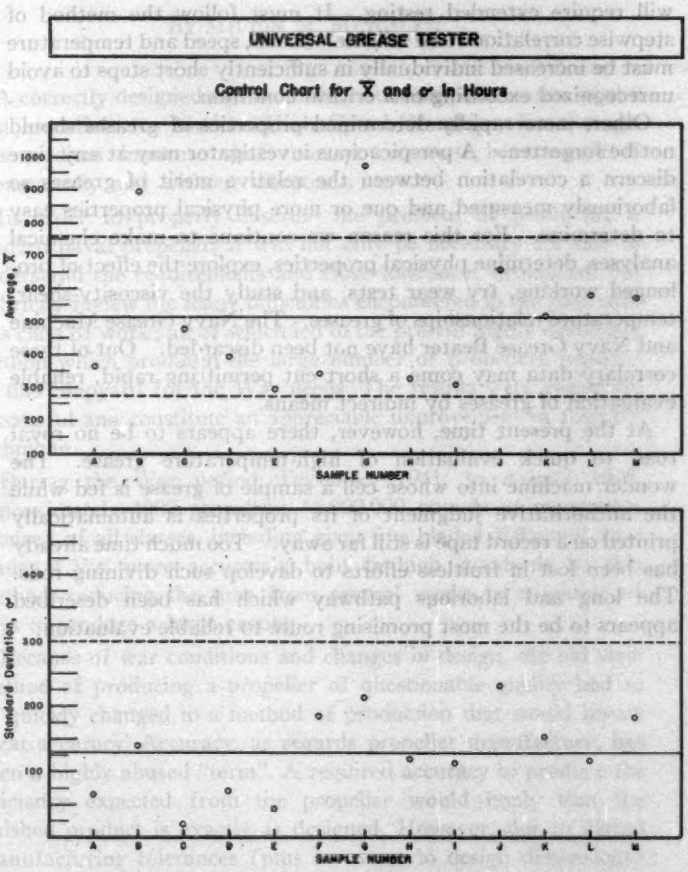


FIGURE 8.—CONTROL-CHART SUMMARY OF TESTS ON APPROVED GREASES.

ing or if another variable, speed or load, should be increased as the next step toward reduction of qualification-test time.

Reduction of the present 4000-hour qualification time for shipboard electric-motor-bearing greases to a few weeks or days will require extended testing. It must follow the method of stepwise correlation to be reliable. Load, speed and temperature must be increased individually in sufficiently short steps to avoid unrecognized exceeding of a critical condition.

Other, more rapidly determined properties of greases should not be forgotten. A perspicacious investigator may at any time discern a correlation between the relative merit of greases so laboriously measured and one or more physical properties easy to determine. For this reason we continue to make chemical analyses, determine physical properties, explore the effect of prolonged working, try wear tests, and study the viscosity-shear-temperature relationships of greases. The Navy Grease Machine and Navy Grease Beater have not been discarded. Out of these corrolary data may come a short cut permitting rapid, reliable evaluation of greases by indirect means.

At the present time, however, there appears to be no royal road to quick evaluation of high-temperature grease. The wonder machine into whose cell a sample of grease is fed while the authoritative judgment of its properties is automatically printed on a record tape is still far away. Too much time already has been lost in fruitless efforts to develop such divining rods. The long and laborious pathway which has been described appears to be the most promising route to reliable evaluation.

MARE ISLAND'S ACCURATE METHOD OF PROPELLER MANUFACTURE.

By Milton M. Metcalf*.

A correctly designed and accurately built propeller is one of the first essentials for Modern High Speed Warships, for through this medium immense shaft horsepower must be transmitted economically and without vibration.

In order to properly describe the problem of producing a modern naval propeller, it will not only be necessary for one to understand the requirements and close tolerances demanded, but to briefly review the many difficulties encountered in the past with this class of work; all of which led to the present highly developed method which produced the large number of propellers required of this Shipyard for the War needs. The results were eminently successful and constitute an appreciable improvement in foundry technique.

During the War period, December 1941, to August 1945, approximately 4000 propellers (6,000,000 pounds of manganese bronze) of all classes, including giant five bladed BB's and four bladed CV's, were accurately built to high standards by this method, reducing the time from several weeks to a matter of days to produce a large casting.

Because of war conditions and changes in design, the old slow method of producing a propeller of questionable quality had to be quickly changed to a method of production that would insure great accuracy. Accuracy, as regards propeller manufacture, has been a highly abused "term". A required accuracy to produce the efficiency expected from the propeller would imply that the finished product is exactly as designed. However, due to liberal manufacturing tolerances (plus or minus to design dimensions) many propellers were produced which, while being theoretically "accurate" did not follow the design close enough to produce the ultimate in efficiency. The "ultimate" as referred to, in the above, is a requirement rather than an exception. Not only are modern

* Chief Draftsman, Machinery, Mare Island Naval Shipyard, Vallejo, California.

propellers of a precise and complex design, but they must be produced under "production line" methods to meet the demand of expanded shipbuilding facilities requiring close time and costs estimates.

One may ask, "What is unusual in expecting production in this field?" The answer is that here we have a product that weighs, let us say, 15 tons. It is large, heavy and unwieldy. Due to its unusual form it is unmachinable except for a highly developed and extremely expensive profiling machine tool of which there is only one in the country. Further, as mentioned before, the product must be finished to a great degree of accuracy.

Mare Island Naval Shipyard developed a method of production that meets all of the modern requirements; a system which utilizes past proven accomplishments together with new and novel methods for rapid production. It is now possible to produce a modern propeller in approximately one quarter the time it took to produce a propeller under old methods, thereby reducing the cost proportionately. It can be appreciated that in a period of time that involved the casting of millions of pounds of manganese bronze into propellers, using this new method, a staggering sum was saved and put to use in other equally important critical fields necessary to the successful culmination of the war.

Early Loam Molding Process.

In order to get a clear picture of the industry from its infancy, let us go back momentarily to the 1890's. At this period the Shipyard was producing propellers by what is known as the loam molding process. The propellers were generally of a design which included true pitch (a helical pressure face) and ogival blade sections. Reciprocating engines with low Rpm. did not demand intricate design and close tolerances of propellers. Very little finish work was done at this period. Castings were of good quality and substantially sound. The men who produced these propellers were excellent artisans and met every requirement of their day.

A pictorial record of Mare Island's early loam molding process, Figures 1 and 3 inclusive taken in the 90's shows this process from the time the pit was dug in the foundry floor (to accommodate the mold) to when the finished casting was uncovered.

They picture the original start of molding with sweep and guide board being used to sweep-up the pressure face of the first blade, the method of molding the blade pattern by using sectional cants and filling between with black molding sand, and the finished casting being removed from the mold.

The loam molding process required a spindle which was set up in the center of the pit and guyed-up with bracing rods to anchors in the sides of the pit. This spindle became the base line for the mold. A guide board built to the proper pitch angle was set in the pit at a point beyond the proposed extremity of the blade mold, and a sweep was installed on the spindle in such a manner as to have the end of the sweep resting on the guide board. Thus, when rotating the sweep on its axis, with the end resting on the guide board, a helix of the desired pressure face form could be produced. The drag of the mold was then built-up to a point a few inches below the desired form with brick; and then a facing of loam added to bring the drag into sweeping distance. The sweep was then moved up and down the guide board producing the pressure face of the blade on the drag half of the mold. This process was repeated for each blade.

After drags were completed, cants (cut to the desired developed cylindrical blade section dimensions and bent to suit the radius of their respective station on the blade) were located in their proper place on the drag faces and the outline of the blades determined with splines. This established the form of the pattern. The intervening space between cants and splines was filled with black molding sand and the exposed surface faired to suit. The result was a molded pattern.

After the pattern was molded, a facing of loam was added. The face of the cope half of the mold was produced in this way. The rest of the cope was built up with brick in much the same manner as the drag. After the loam had set, the copes were lifted and the blade and hub patterns were removed. After the copes and drags were dried and cured with heat, they were assembled, hub core put in place, various necessary pouring equipment located, and heat was started on the bronze material.

The above process was long, costly and did not produce accuracy in the casting.

Pre-War Method of Molding with Wooden Pattern.

About 1906 wooden propeller patterns were in general use. This change was brought about by the need for a more uniform casting. A set of design tolerances was established by the Bureau of Steam Engineering (BuShips) which required the propeller pressure face to be machined to a true helicoidal surface, and more attention to be devoted toward producing a finished product. This was necessary due to the advancement in propulsion machinery which included higher steam pressures, direct drive turbines, and high Rpm. In order to machine the pressure faces of the blades, it was necessary to allow an excessive amount of finish on the pattern to make up for the deformation due to shrinkage and mold shifting. This amount of shrinkage added to the pattern still kept the casting from being classed as an "accurate" casting due to the amount of metal to be removed before the finishing process was completed. Also, there was no particular advancement in the process other than the change to wooden patterns and the introduction of molding sand instead of loam. The pressure faces of the blade molds were still being swept up with the sweep and guide board method.

Figures 4 to 6 (inclusive) are views of this later process and illustrate the differences in methods over the loam process, but still show the similarity that existed. It will be noted that conditions related to the molder himself show no improvement whatsoever. The very nature of the process allowed for no better working conditions. It is assumed that this group of pictures needs no further explanations other than the description given in the following paragraphs.

As is noted above, the change in process to the above mentioned method has not advanced much over the former process other than to produce uniformity in the casting. The same type of pit had to be dug in the foundry floor as before. The pressure faces of the blade molds were still being swept up, and other related equipment still designated the job into the "piece-work" class. No "production-line" could be maintained by this process. Unique in itself was the method used to hold down the mold, after it was assembled, in an effort to keep it from shifting. The usual practice was to place a weight (equal to 3 times the estimated weight of the rough casting) on top of the mold. This made a huge pile of

assorted weights, lead blocks, blooms, girders and beams that constituted a hazard to men engaged in the pouring process. Men with puddling rods inserted in the riser over the hub were required to work over the top of this complicated rig.

This process produced castings of "high grade" chemical and physical properties. There was a great uniformity in the blades and the results were very desirable under requirements of that time. Still, when these castings were machined, the best part of the casting (or fine grained chill surface) was removed. It was noted, that in many cases, where a uniform deformation had been obtained from shrinkage, the machine tool set to cut the desired pitch would cut off about ¾ of an inch of material on one side of the pressure face of the blade, while barely cleaning up the other side of the same face. Naturally this same condition would appear on the suction faces which had to be chipped into shape, and as was the custom at that time, the suction faces did not get much attention. They were generally left pretty rough from an accuracy point of view. The requirement that propellers be dynamically balanced and the subsequent removal of unbalance weight from the suction faces of the blades further aggravated this condition.

A summary of the above conditions reveals an overall extravagant waste of time, labor and material; an expensive product to finish; an antique method of producing that could not lend itself to mass production, and, finally, a finished propeller whose inaccuracies could not be tolerated for modern exacting requirements.

Present Accurate Method on Production Basis.

As mentioned in the opening paragraphs of this article, modern design and accuracy necessitated changes in the manufacturing procedure. The new design eliminated the old ogival section with helicoidal pressure face. This was replaced by the more efficient air-foil or hydro-foil type of section as in the standard "Clark-Y" or the newer "Mean Camber Line" sections. These newer propellers also complicated manufacturing methods by having double camber faces (on many of the designs) added to increase the strength. Experiment with these new sections indicated that a greater percentage of the overall efficiency of the propeller de-

pended upon the condition of the suction face (back) of the blades. This upset the theory that contended that major part of the work was done by the pressure face.

Here was a challenge to the industry! A necessary change in design had placed a propeller blade in a non-machinable category except for the profiling machine mentioned earlier in this article. War time conditions could not permit the manufacture of this machine or allow an excessive amount of hand finishing to these propellers. This Shipyard was required to undertake production of propellers at an unbelievable rate of production. This production was not to be confined to "type" designs (as it was with some shipyards) but designs that were to change with conditions and other criteria.

As far as was known at this Shipyard, the only attempt at mass production of large warship propellers prior to this war was made by the Thatcher Propeller and Foundry Corporation of Albany, New York. This process is thoroughly described in the February 1922 issue of the JOURNAL. This method was apparently effective in producing large numbers of like propellers under old standards of design and accuracy. It was felt that much improvement was desired over this method and that something more flexible and modern could be developed.

In 1939, the author, while on a visit to the Bureau of Ships, proposed and received the "go ahead" to produce accurate propeller castings with modern equipment, under modern working conditions, that would require no machining other than hub boring, facing, drilling and minimum amount of chipping and grinding to finish within closely maintained tolerances.

It was proposed to construct in the foundry a concrete pit thereby eliminating the need of digging a separate pit for each job. This pit was to contain a heavy circular, machined floor plate accurately set therein to maintain a perfectly level and face plate surface. Cast iron cope and drag boxes were to be made to accommodate several different diameters and pitch characteristics. These boxes were to have machined bases and mating flanges. They were to be interchangeable; that is, any cope would fit any drag for the same type of mold assembly. Actual molding was to take place on another floor plate, outside of the pit, on the foundry floor. Guides and dowels were to be used to insure

perfect placement in both the molding plate and pit plate. Both copes and drags were to be dried and cured in the ovens to insure uniform characteristics in both units. Mold boxes were to feature rigidity, strength, simplicity and accessibility for operation. By bolting the drags and copes together, and this assembly to the circular floor plate, it was possible to eliminate the great weight of material previously used to hold down the mold.

The all important advancement made was the introduction of a hand finished metal pattern. This pattern was developed by first making a wooden pattern with enough finish to compensate for shrinkage. The pattern was to be then cast of manganese bronze and the actual shrinkage plotted and used as a basis for finishing the metal pattern. The result of this was a single bladed pattern with a bastard pitch and variable thickness shrinkage allowance.

Another feature to be considered in the design of this equipment was the need to make the process fool-proof, so as to utilize a supply of inexperienced molders thereby releasing the more skilled mechanics for other necessary operations in the Foundry. Naturally, the main object behind all of this was to diminish the time required to produce a casting. It was estimated that a savings in time of 75 per cent would be realized per propeller.

The first wheels produced by this method were a group of propellers for the "AVP10 Class" of vessels. These were 3-bladed propellers, 9'-2" in diameter. They had hydrofoil sections, double camber faces and variable pitch. The results of the first wheel cast were beyond all expectations, as this casting was practically to finished size. It was decided that henceforth, all propellers cast by this method were to have a finish allowance of only 1/32" on the pressure face and 1/16" on the suction face of the blades.

A series of pictures showing this new method are shown as Figures 7 to 21 inclusive. Figure 7 pictures the metal pattern developed from the original wooden pattern and finished to provide shrinkage correction. Figure 8 shows the ramming of drag box on molding plate. The pattern is held rigidly by both spindle stud and strongback to prevent pattern blade from raising under the ramming process. Figure 9 shows cope box being fitted to finished drag and Figure 10 shows ramming of cope. Figures 11 and 12 show completed copes and drags awaiting assembly after delivery from drying oven. Figure 13 shows assembly of drags on casting floor plate in pit, while Figure 14 shows copes secured

to drags with pouring trough in place. It is noted that two gates were used on this casting. Figure 15 shows pouring operation. Note lack of excess rigging and staging as compared to previous pouring methods, everything about the mold being accessible for examination and inspection. Figure 16 shows rough casting removed from mold with riser still intact but gates removed.

Figures 17 to 20 (inclusive) are a set of photographs showing "plus or minus" figures (decimals of inches) on the pressure faces of the blades of the rough casting shown in Figure 16. These figures (dimensions shown) indicate the amount of metal to be removed to bring that face of the blades to the designed dimensions. These plus or minus figures are determined from a chart prepared by the Machine Shop from a special layout for that purpose. Readings are taken with a precision pitchometer for convenience. The layout as shown in this group is preliminary, yet shows to a large degree, the accuracy obtained by this casting method. For finishing layouts, checks are made on the accessible sectional radii corresponding to those given on the plans and at every station on these radii that fall on 5° spacings on each side of the designed centerline. Figure 21 shows the finished propeller awaiting the final polish, metal edge binding and delivery to store. Figure 22, a recent photograph showing a group of late design Destroyer propellers, further illustrates the accuracy of this casting method. This picture shows a rough casting with riser intact, a semi-finished propeller and a finished propeller. It is noted that the semi-finished wheel has the suction face drill spotted on upper blades, reference grooves chipped and partially surface-chipped on lower blades.

It can be appreciated that the biggest step toward mass production has been taken in the foundry. A job that formerly took weeks to accomplish can now be done in a matter of a few days or shifts. Besides speeding up production, the accuracy of the casting has lowered the finishing time to a minimum without sacrificing the quality.

The first step of the finishing process is to chip off the individual blade edge risers and then to make a preliminary layout on the rough casting. From this layout, the propeller proceeds to the boring mill where the big riser is removed and the hub is bored and faced. Next, the key ways are cut and all drilling and

PLATE I.

FIGURE 2.

FIGURE 3.

FIGURE 4.

FIGURE 5.

FIGURE 6.

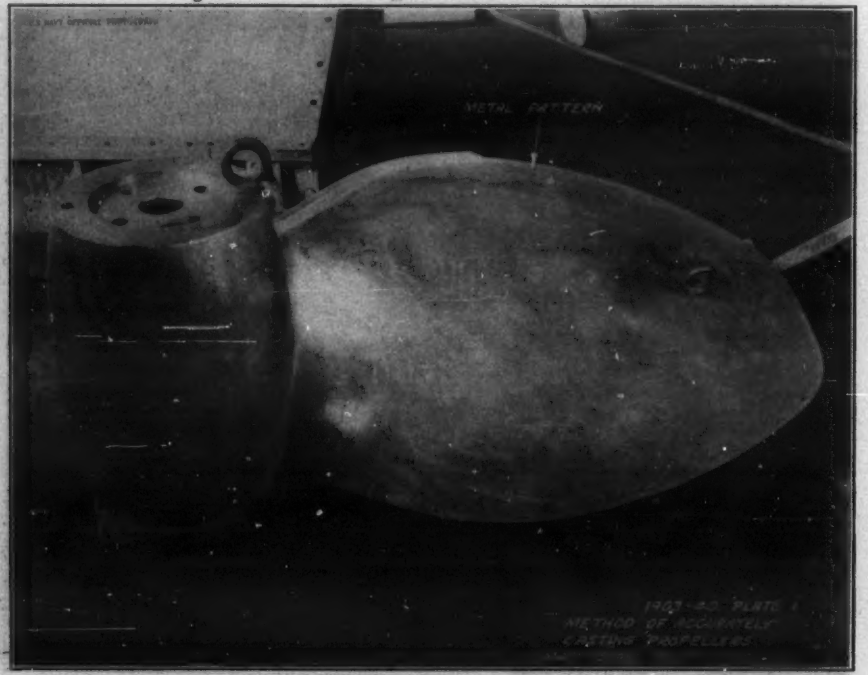


FIGURE 7.

FIGURE 8.

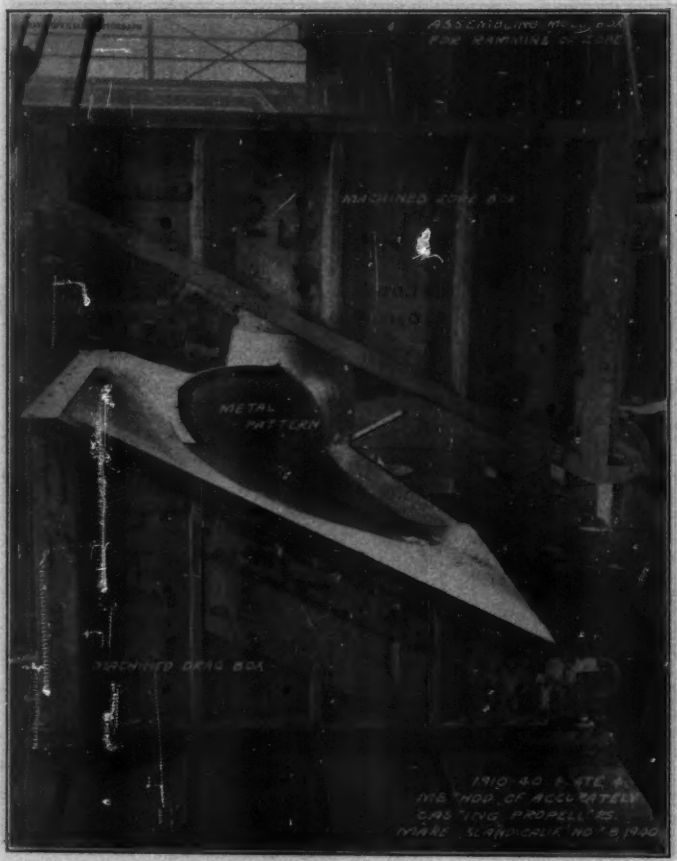


FIGURE 9.

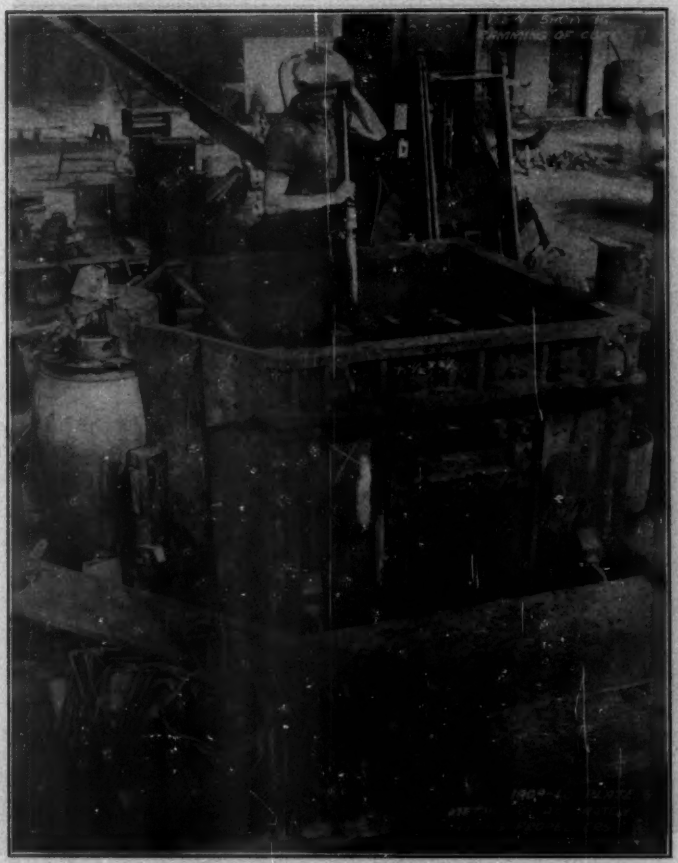


FIGURE 10.

FIGURE 11.

VIEW OF FINISHED CORE
HALVES READY FOR ASSEMBLY
ON CASTING PLATE
METHOD OF ACCURATELY
CASTING PROPELLERS

FIGURE 12.

VIEW OF DOG MOLDS
ASSEMBLED IN PIT
VIEWING OF ACCURATELY
CASTING PROPELLERS

FIGURE 13.

VIEW OF ASSEMBLED MOLD
POURING TROUGH IN PLACE
READY FOR POURING

FIGURE 14.

FIGURE 15.

FIGURE 16.

FIGURE 17.

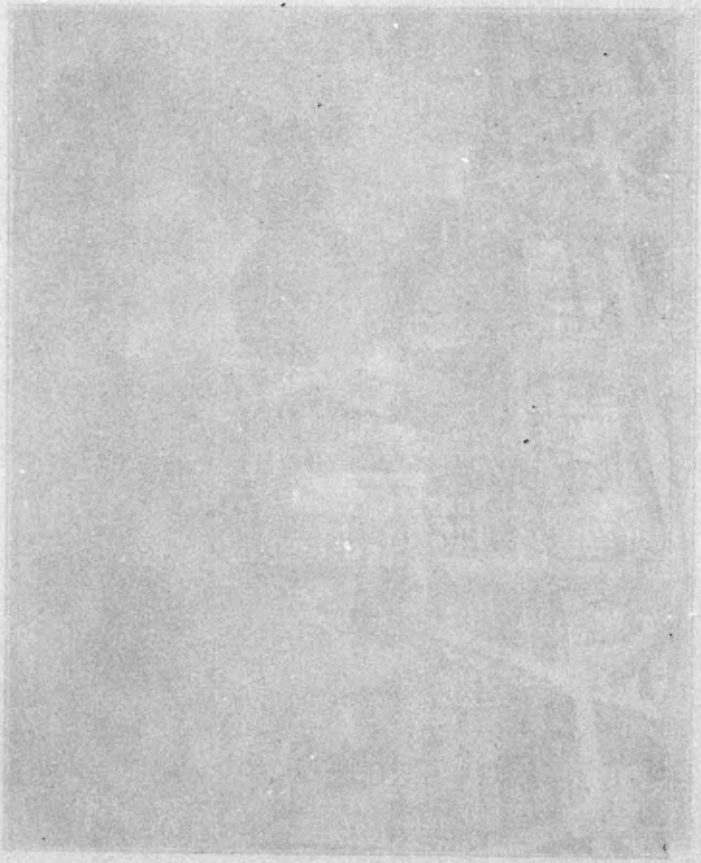
FIGURE 18.

+.010
.000
+.030
+.050
+.070
+.090
+.110
+.130
+.150
+.170
+.190
+.210
+.230
+.250
+.270
+.290
+.310
+.330
+.350
+.370
+.390
+.410
+.430
+.450
+.470
+.490
+.510
+.530
+.550
+.570
+.590
+.610
+.630
+.650
+.670
+.690
+.710
+.730
+.750
+.770
+.790
+.810
+.830
+.850
+.870
+.890
+.910
+.930
+.950
+.970
+.990
+1.010
+1.030
+1.050
+1.070
+1.090
+1.110
+1.130
+1.150
+1.170
+1.190
+1.210
+1.230
+1.250
+1.270
+1.290
+1.310
+1.330
+1.350
+1.370
+1.390
+1.410
+1.430
+1.450
+1.470
+1.490
+1.510
+1.530
+1.550
+1.570
+1.590
+1.610
+1.630
+1.650
+1.670
+1.690
+1.710
+1.730
+1.750
+1.770
+1.790

FIGURE 20.

FIGURE 21.

FIGURE 22.



tapping on the hub accomplished. The next step is to complete the layout on the pressure face of the blades and drill spot these faces for chippers and grinders reference. The pressure faces, including the fillets on this side of blade, are then finished to size. Fillets are checked with metal templates. The propeller is then turned over and, using the completed pressure face as a reference, a thickness layout is made on the suction faces of the blades by use of special transfer calipers. The suction faces are then drill spotted and chipped together with the fillets on this side of the blades. Blade edges are chipped to edge templates. Entire backs of the blades are then ground and polished. The propeller is then again turned over and a pitch check made to determine if chipping the backs has in any way distorted the pressure face. The propeller is then dynamically balanced in the Akimoff machine. It is here noted that, under normal conditions, very little metal is required to be removed to effect dynamic balance.

After balancing is completed a final pitch and thickness check is made. This check is recorded and becomes a permanent part of the Shipyard records on this particular propeller. The finishing process has developed into a highly specialized trade and requires the best efforts of the mechanics involved. A short while on the job usually brings out the best in the man wherein he builds up a pride in his work not usually found in many other occupations.

Templates and full sized layouts used by the Machine shop to prepare their working charts in conjunction with the pitchometer measurements are prepared by men in the Mold Loft. This utilizes a source of experienced men highly trained in descriptive geometry.

The Pattern Shop, whose specialized work supports and is the backbone of the whole propeller manufacturing structure, has developed a patternmaking system that, it is felt, cannot be matched throughout the industry. Specialized equipment has been developed for generating the contour of the pattern blades. A system of making the pattern hub allows for a turned surface, and the method of inserting the fillets after the hub and blade of the pattern is finished insures accurate fillets on the casting.

Another control in the Foundry, which has not been mentioned before, is the services of a metallurgist, who, working in con-

junction with the Foundry provides the required metallurgical advice. It is his duty to maintain the quality of the cast material and to keep a record of the physical properties of the test bars attached to the propeller blades. This record enables the Foundry to maintain the quality that is expected of them.

The Design Section of the Shipyard provides the engineering consultants necessary to aid the progress of the plant and insure a positive interpretation of the plans. It is the Design Section's aim to encourage the ideas of the man within the propeller field and to further their interests by assisting them to incorporate their ideas into the method of production. Past ideas presented by the men in the shops have paid big dividends and in some cases changed the basic principles of the molding methods. All of this has led to a better product and has saved thousands of dollars in time and labor by adoption of these ideas alone.

DRY TYPE DISTRIBUTION TRANSFORMERS FOR SHIPBOARD POWER AND LIGHT.

By J. H. Chiles, Jr. and Paul R. Sprowl*.

Previous to World War II, the Navy had plans under way for more extensive application of A.C. power supply on combat vessels. Transformers had first been installed on the U.S.S. *Quincy* (CA39) commissioned in June, 1936. By 1938 formal purchase specifications had been formulated and designs made available for dry type distribution transformers through 50 KVA, single phase. Original Navy designs departed but slightly from standard commercial designs. In general, the only differences were the special voltage ratios (450 volts primary to either 117, 120 or 230 volts secondary); a special, rather complicated terminal arrangement for delta connection of primary and secondary; and the elimination of cast iron or aluminum end frames. Figure I shows a transformer typical of the 50 KVA design which weighed 1050 pounds. The stimulus of wartime development has reduced this weight to 447 pounds (see Figure V).

In 1940 the Navy found a considerable increase in the amount of anti-aircraft armament and other topside equipment to be desirable. Installation of such equipment, however, increased materially the topside weight of vessels, thus tending to reduce their stability. In view of this fact, a reduction in the weight of equipment installed in vessels, particularly that located above the water line, was found necessary. Added emphasis was therefore placed on the reduction of the weight and space requirements of naval electrical equipment. This fact, plus the increasing demand for transformers due to expanding shipbuilding work, appeared to justify a development program for transformers. Accordingly, the transformer manufacturers, in response to a request from the Bureau of Ships, undertook the development of improved transformers designed to meet the more stringent requirements. Several additional non-standard KVA ratings (20, 30 and 60 KVA) were added to the transformer specifications at this time, thus providing finer weight and volume increments;

* Mr. Sprowl was formerly with U. S. Navy Electrical Section, Bureau of Ships, Washington, D. C.—Both from Transformer Div., Westinghouse Electric Corporation, Sharon, Pa.

and the range of ratings was extended through 100 KVA to meet the demands of increased loads.

The new designs were characterized by reduced volume, reduced weight (weighing approximately 80 per cent of previous designs), and increased mechanical shock resistance. Figure II shows the new 1941 light-weight 50 KVA design which used Class A insulation (55° C. rise organic materials), and which resulted in a weight reduction from 1050 pounds to 840 pounds.

In addition to the above characteristics, an entirely new case construction included a terminal chamber designed to eliminate the complex terminal board previously used. This terminal chamber was so designed that the transformers could be installed in three phase delta-delta banks with the three-conductor high-voltage and low-voltage ship's cables entering through drilled end-cover plates. Clamping blocks of laminated phenolic were provided inside the connection compartment for supporting and securing the ship's cables. (See Figure III.) Connections between the ship's cable and transformer coil leads were made using tee-type solderless connectors which could be readily disconnected in the event it became necessary to remove a damaged transformer from service. Open delta connection providing 58 per cent of the bank rating could be established simply by disconnecting the leads of the damaged transformer.

By 1942, the technique of application to transformer designs of Class B insulating materials* had progressed to the point where it was as economical to design transformers with Class B materials as with Class A materials. At a number of conferences, the Navy requested that development of new designs based on Class B materials be considered in order to obtain further size and weight reductions. In addition to weight and size reductions over previous designs, it was requested by the Navy that the new transformers be designed to withstand the much more severe shock test of 2000 foot-pounds instead of the previous level of 150 foot-pounds which had been found to be inadequate as a result of battle experience. It can be readily understood that these conflicting requirements presented quite a problem.

New core materials came into use at this time. These materials, with the same losses, permitted reducing the core weight and copper weights materially. These reductions were

* Class B materials are inorganic materials such as mica, glass, asbestos, etc., which were at this time rated at 75° C. rise.

G-218

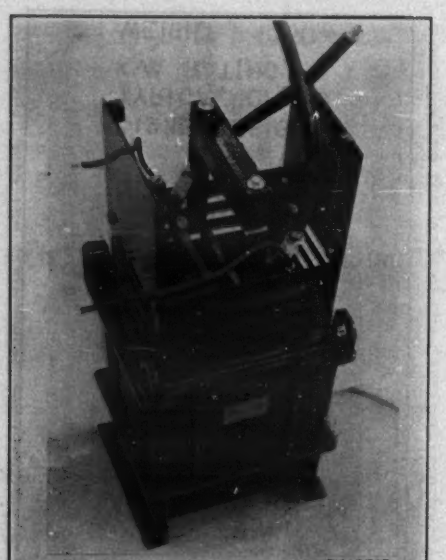
FIG. I

1938 DESIGN
TYPICAL OF 50
KVA RATING
WEIGHT 1050#

SH-6823

FIG. II

1941 LIGHT WEIGHT DESIGN
TYPICAL OF 50 KVA RATING
WEIGHT 840#



SH-6315

FIG. III

1941 DESIGN SHOWING SHIP'S CABLE CLAMPS AND REMOVABLE END PLATES TO BE DRILLED AT INSTALLATION FOR SHIP'S CABLES

SH-8333

FIG. IV

1942 CLASS B DESIGN TYPICAL OF 50 KVA RATING WEIGHT 580#

SH-10064

FIG. V

1944 DESIGN TYPICAL OF 50 KVA RATING WEIGHT 447#

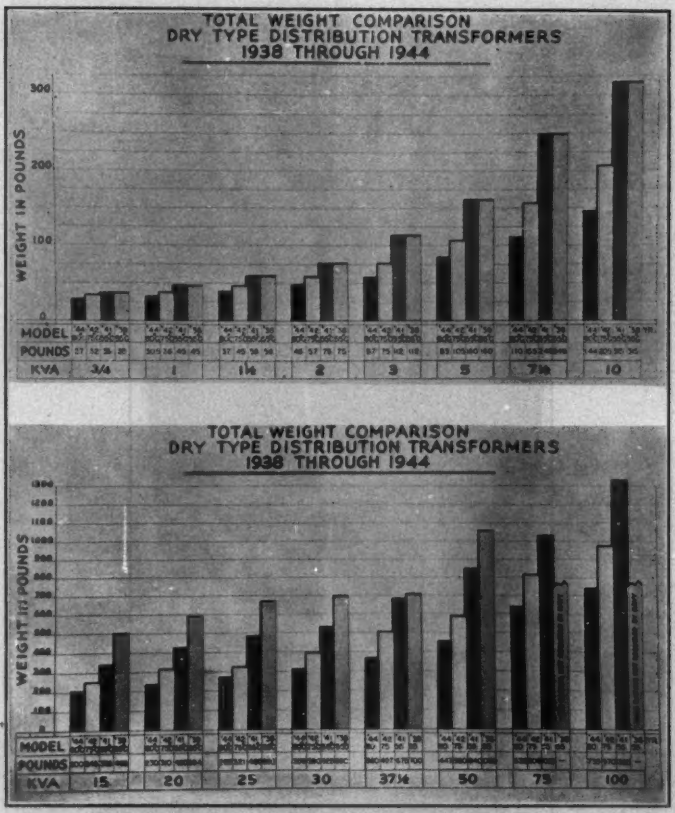


FIG. VI

DESIGN PROGRESS CHART SHOWING WEIGHT REDUCTION FROM 1938 THRU 1944

effected because the better performance of the new core steels permitted a reduction in the core weight with a corresponding reduction in the coil and case weights. Figure IV. shows a 50 KVA 1942 design weighing 580 pounds compared to the previous unit weighing 840 pounds.

While it would appear that most remarkable progress had been made in weight and size reduction, nevertheless in the latter part of 1943 and the early part of 1944 some manufacturers and the Navy felt that by the use of data accumulated from designs then in production further improvements could be made. In the meantime, American Standards Association had recognized and established the limits of Class B insulating materials as 80° C. rise instead of the previous 75° C. value. The Navy felt that this increase in operating temperature, plus other developments in design, would warrant a redesign of the complete line. Typical preliminary designs indicated that further reductions in weight of 20 to 25 per cent could be made.

At this point, the Bureau of Ships felt that a "Navy Distribution Transformer Committee" should be organized, in order to obtain not only the lowest possible weight, but also to study other problems such as case construction, simplification of connections, reduction of magnetizing inrush currents, and particularly to attack the seemingly impossible problem of making the designs of all manufacturers interchangeable in regard to mounting dimensions.

The objective of establishing interchangeable mounting dimensions appeared to be particularly difficult to attain, if not impossible, for in addition there were the problems of materially reducing weight and volume, meeting the stringent specification requirements on losses, exciting current, regulation, inrush current limitations, and shock strength. For a number of years, due to the emphasis of light weight and material saving, there had been keen competition between manufacturers to maintain a weight and volume advantage well within the specification requirements, and there was a natural reluctance to give up any such advantage as might be lost by restrictions on mounting dimensions. Furthermore, some of the Committee members were familiar with some of the difficulties encountered in standardizing certain dimensions of other relatively simple types of apparatus in industry; and with the large number of ratings

involved, the very stringent weight and shock requirements, the Committee realized it was faced with a most difficult problem. It presented a real challenge that required the utmost of cooperation between representatives on the Committee, as well as a great deal of very intensive design work to make sure that the Committee's recommendations could be met.

Despite the above difficulties, the Committee entered on its assignment with such complete cooperation that within a period of six to eight months, complete recommendations had been made to the Bureau on the following:

(1) Drastic reductions in specification weights and volumes.

(2) Due to the materially smaller physical size, the omission of cable clamps in the cable connecting compartments was recommended.

(3) A reduction in the number of different KVA sizes from 17 to 11 as a result of an analysis of the activity of the various ratings.

(4) An extension of bulkhead mounting from the former upper limit of 10 KVA to an upper limit of 25 KVA. This permits a considerable saving in the weight of the foundation formerly required for deck mounting of the 15, 20 and 25 KVA ratings.

(5) A recommendation for mounting dimensions for each of 17 different KVA sizes, including center to center spacings between units, so that any manufacturer's design would be interchangeable with any other design built to the Navy Specifications.

As previously pointed out, item (5) in the above recommendations is of particular significance in that it required a great many compromises and concessions which in some cases would seemingly penalize one or another manufacturer in weight or volume due to his particular design practice. As an example, in certain ratings some manufacturers used the shell form of core and coil construction, while others used the core form. Obviously, there needed to be a close mutual cooperation as to construction methods to withstand shock in order to arrive at common mounting hole dimensions between manufacturers. In some cases the close limitations in space and weight required a great deal of ingenuity in order to arrive at a Committee agreement so that the varying design practices could be fitted into standardized dimensions without penalizing weight, volume, or the factor of

safety for shock strength. Figure V shows a 50 KVA rating typical of the new design.

The fact that wartime emphasis can stimulate developments which result in material savings is amply evidenced by the Chart in Figure VI. This chart shows almost a year by year saving in weight of a vital, and one of the most reliable components of the electrical system of modern ships. Certain illustrations of the weight reductions are of interest; for example, the specification in 1938, permitted 1300 pounds weight for a 50 KVA transformer. A typical design at that time weighed 1050 pounds, or 81 per cent of the specification; while the 1944 design weighed 447 pounds, which is 43 per cent of the original 1938 design, or 34½ per cent of the original specification value in 1938. Similarly, the 15 KVA design in 1938 was 92½ per cent of the specification limit. In 1944 its weight was 37½ per cent of the 1938 specification limit, and 40½ per cent of the 1938 design. There are few electrical components that can boast of such a trouble-free history of wartime service, with so stringent an increase in service requirements, and with an overall weight reduction to the order of 40 per cent of its previous value. Figure VII gives perhaps a better illustration of the progress over six years, that is 3½ times the KVA for 90 per cent of the weight.

FIG. VII

DESIGN PROGRESS CHART SHOWING 3½ TIMES THE KVA WITH ONLY 90% OF THE WEIGHT

YEAR	1938	1941	1942	1944
KVA	15	25	37½	50
WEIGHT	495#	480#	497#	447#

This program of weight reduction shared by a number of manufacturers, along with the standardization program of the Navy Dry Type Distribution Transformer Committee, is truly a challenge to other sections of Industry.

THE FUTURE OF THE HIGH TEMPERATURE STEAM PIPING EXPANSION PROBLEM.

By F. A. Scanlan*.

Considerable current engineering literature is being devoted to two significant phases of power generation of special interest to marine and naval engineers, the development of the gas turbine and material research for high temperature steam piping.

In the field of the gas turbine, materials have already been developed to such a point as to allow the actual testing of a unit of 3500 horsepower with a maximum temperature of 1300° F. This, of course, refers to recent work by the Bureau of Ships at Annapolis, where it is understood that further tests to the 1500° F. design limit of the materials are expected to be conducted. Other test installations, here and abroad, are known to be under development, some constructed to withstand temperatures as high as 1600° F. to 1800° F. in order to obtain maximum thermal efficiency. The largest unit constructed to the present is one of 10,000 K.W. in Europe[1]. The prospect of very high thermal efficiencies for gas turbine plants has been recently tempered by the realization that the power consumption of the necessary auxiliaries must be charged to the main unit as in steam practice. Reported fuel consumption, approaching 0.45 lbs. of fuel per shaft horsepower per hour[2], is an indication as to what can be expected of plants using temperatures of about 1300° F. and having capacities around 2500 horsepower. With increasing size of units and use of higher temperatures, betterment of this fuel consumption should be realized; but the period of overenthusiasm based entirely on theory, has passed. It can be deduced from data thus far that this type of thermal prime mover will, in the immediate future, find its greatest field in units of sizes from 2000 horsepower to 15,000 horsepower as a competitor to large Diesels and small steam engines and turbines. It has already proven useful in certain chemical and industrial processes where large amounts of compressed air are required, rather than high efficiency. Great credit must be given also for the stimulus in

* Member, Boston Naval Shipyard.

research in high temperature material brought about by the development of the gas turbine.

On the other hand, steam turbine installations with specific fuel rates in the vicinity of 0.50 pounds[3] per shaft horsepower per hour at pressures varying from 1200 psi to over 2000 psi with a maximum temperature of about 950° F. have been reported for both vessel propulsion and shore station plants. Some of these installations are comparatively simple, using neither reheat nor any overcomplication of regenerative feed heating. These plants range in size from 8500 shaft horsepower up. Thus, it seems that the incentive to increase the efficiency of steam plants for the purpose of reducing fuel consumption is attractive. Also, in view of the performance of the steam turbine under conditions imposed by World War II, any successful step toward improved economy will help to insure the future availablity of this flexible and rugged prime mover.

For many years, the exploitations of the use of very high temperatures for steam power plants was prevented, particularly by the lack of a satisfactory piping material which would resist "creep" above 750° F. A partial solution to the problem was brought about by the development of a carbon steel piping material having about ½ per cent molybdenum, which has permitted comparatively recent use of temperatures up to the vicinity of 900° F. Lately, it has been discovered that exposures of this CMO piping to temperatures above 900° F. subjects the welds in such piping to what is known as "graphitization." This complicated process results in the loss of carbon from the steel, exposing the structure to brittle failure. This tendency has been noted at temperatures as low as 850° F. Sufficient laboratory work has been done already which promises a satisfactory solution to the "graphitization" problem. The proposal involving the reduction of carbon content to the practicable minimum of less than 0.05 per cent, without loss of "creep" properties for CMO, is encouraging[4]. As a result of gas turbine work, we are also fortunate in having knowledge of nickel alloys of types, such as "Inconel" and "K" monel, which have excellent "creep" properties, comparable to the alloy steels that are used for the units of high temperature piping systems, and which are at the same time capable of fabrication. It is to be noted that in gas turbine work, ducts or piping, while being subjected to the very

high temperatures of the operation, are not required to withstand internal pressures of more than 100 psi up to the present. Ultimately, it is to be expected that pressures of as high as 850 psi will be encountered. It is also possible to protect gas turbine ducts with insulation installed between an inner sleeve and the outer duct. It is conceivable that it will be necessary in the future to resort to this type of piping for very high steam temperatures.

While we know that the greatest theoretical gain in the efficiency of the steam plant can be accomplished by carrying the regenerative cycle to the ultimate with existing pressures and temperatures, such design at present would not be feasible for shipboard installation where space and weight are at a premium. If we study the temperature entropy diagram and the Mollier chart, keeping in mind the pressures and temperatures now in use, the desirability of going into higher temperatures is evident. We also know from experience that the theoretical gain from superheating has been accompanied by a substantial benefit in turbine operation. Hence, any successful effort in the use of higher temperature steam can be expected to be satisfactorily rewarded, and it is logical to take this step, considering the foregoing.

Before applying new materials for high temperature steam piping above 850° F. to 900° F., for which sufficient "creep" data is available, a careful study of existing design should be made in order that such materials will be economically and intelligently used. Alloys withstanding any temperatures above this range will be not only more costly and less abundant than those presently in use, but fabrication techniques will be required which must be capable of very close control in order to reduce, as much as possible, difficulties similar to those that have occurred in the past when an endeavor was made to use increased temperatures. Of the major problems requiring special consideration for the increasing temperature, only one, namely that of caring for expansion, will be considered in this article.

Up to the present, loops and bends have successfully provided a solution to the expansion problem while keeping the combined stresses within the permissible limits of the available material. With the employment of higher steam temperatures and the consequent reduction in the available strength of piping materials,

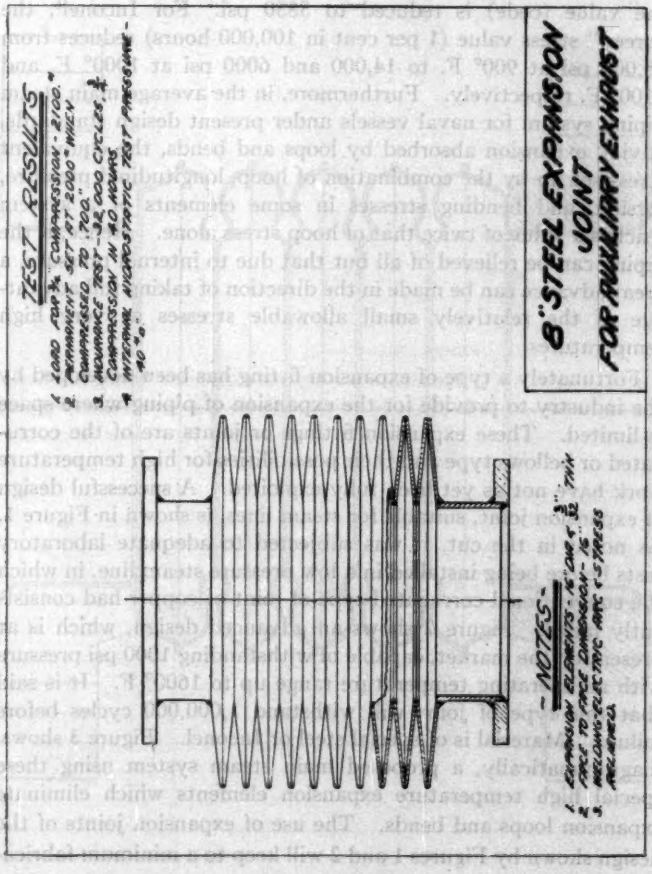

8" STEEL EXPANSION JOINT FOR AUXILIARY EXHAUST
TEST RESULTS
1. 1000 PSI % COMPRESSION - 2100 CYCLES
2. PERMANENT SET AT 2000 CYCLES WHEN COMPRESSED TO 20%
3. ADVANCE TEST - 32,000 CYCLES, 4% COMPRESSION IN 40 HOURS
4. INTERNAL HYDROSTATIC TEST PRESSURE - 90 #.
NOTES
1. CORRUGATED ELEMENTS - 16 GA. STAINLESS STEEL
2. FACE TO FACE DIMENSION - 16"
3. WELDING ELECTRIC ARC STRESS RELIEVED.
3/16"
3/16"

FIGURE 1

serious consideration must be given to finding means other than the above for caring for the expansion. The importance of this will be recognized in the fact that for plain carbon moly steel piping, the codes and specifications allow a stress value of 11,200 psi to 15,000 psi for a temperature of 850° F., while at 1000° F. the value (code) is reduced to 5850 psi. For Inconel[5], the "creep" stress value (1 per cent in 100,000 hours) reduces from 28,000 psi at 900° F. to 14,000 and 6000 psi at 1000° F. and 1100° F. respectively. Furthermore, in the average main steam piping system for naval vessels under present design standards, having expansion absorbed by loops and bends, the equivalent stress set up by the combination of hoop, longitudinal pressure, torsion, and bending stresses in some elements of a system reaches a value of twice that of hoop stress alone. Hence, if the piping can be relieved of all but that due to internal pressure, a great advance can be made in the direction of taking full advantage of the relatively small allowable stresses at very high temperatures.

Fortunately a type of expansion fitting has been developed by the industry to provide for the expansion of piping where space is limited. These expansion fittings or joints are of the corrugated or bellows type and their possibilities for high temperature work have not as yet been fully exploited. A successful design of expansion joint, suitable for steam lines, is shown in Figure 1. As noted in the cut, it was subjected to adequate laboratory tests before being installed in a low pressure steam line, in which the conventional corrugated type of joint of copper had consistently failed. Figure 2 shows an advanced design, which is at present on the market, capable of withstanding 1000 psi pressure with an operating temperature range up to 1600° F. It is said that this type of joint will withstand 1,000,000 cycles before failure. Material is of special steel or Inconel. Figure 3 shows, diagrammatically, a proposed main steam system using these special high temperature expansion elements which eliminate expansion loops and bends. The use of expansion joints of the design shown by Figures 1 and 2 will keep to a minimum fabrication problems involved in the manufacture and installation of a high temperature steam piping ststem. If materials to be used for the other elements of the system are highly sensitive to welding and heat treatment, connections of the purely mechanical

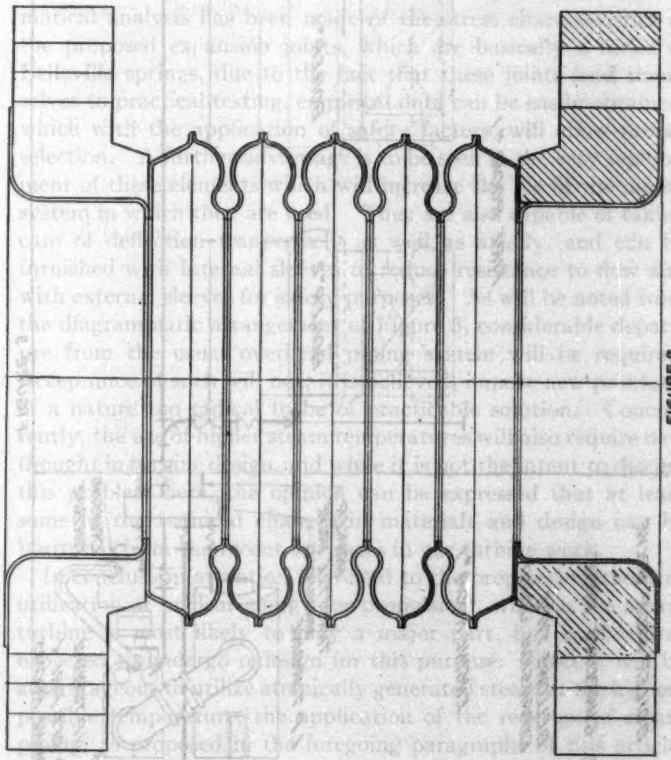


FIGURE 2

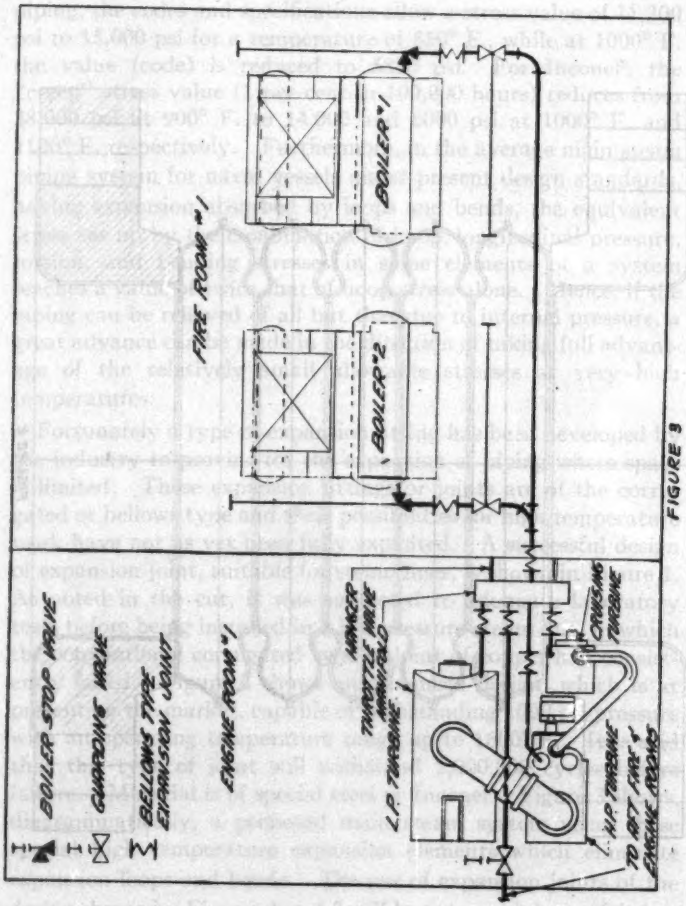

BOILER STOP VALVE
GUARDING VALVE
BELLOWS TYPE EXPANSION JOINT
FIRE ROOM #1
ENGINE ROOM #1
BOILER #1
BOILER #2
MANOEUVRING THROTTLE VALVE
L.P. TURBINE
CRUISING TURBINE
H.P. TURBINE
TO AFT FIRE & ENGINE ROOMS

FIGURE 3

type may be resorted to, thereby eliminating the hazard involved in welding some of our complicated alloys. The trusting of work, which is of laboratory level, to the mechanical personnel involved in fabrication and installation in shop and field would then be obviated. The proposed joints are of such character that they can be carefully fabricated, treated and tested under strictly technical conditions. While no satisfactory rational mathematical analysis has been made of the stress characteristics of the proposed expansion joints, which are basically a series of Belleville springs, due to the fact that these joints lend themselves to practical testing, empirical data can be easily obtained, which with the application of safety factors, will allow proper selection. A further advantage is to be seen in the easy replacement of these elements which will increase the life of the piping system in which they are used. They are also capable of taking care of deflection transversely as well as axially, and can be furnished with internal sleeves to reduce resistance to flow and with external sleeves for safety purposes. As will be noted from the diagrammatic arrangement of Figure 3, considerable departure from the usual overhead piping system will be required. Acceptance of such will not, it is believed, impose new problems of a nature too radical to be of practicable solution. Concurrently, the use of higher steam temperatures will also require new thought in turbine design, and while it is not the intent to discuss this problem here, the opinion can be expressed that at least some of the required changes in materials and design can be borrowed from the recent advances in gas turbine work.

In conclusion, attention is invited to the proposed methods of utilization of nuclear energy for propulsion[6], wherein the steam turbine is most likely to play a major part, but is of course expected to undergo redesign for this purpose. Since it will be advantageous to utilize atomically generated steam at the highest possible temperature, the application of the redesign of steam piping, as proposed in the foregoing paragraphs of this article, will aid materially in approaching the great power developments of which we are now upon the threshold.

References.

1. "Power", September 1946, Page 104.
2. "A Marine Gas Turbine Plant", Soderberg, Smith & Scott "A.S.N.E. Journal", February 1946.

3. "Marine Engineering & Shipping Review", January 1946, Page 147—Discussion by W. W. Smith.
4. "The Welding Journal", January 1946—"Graphitization" —George A. Timmons.
5. International Nickel Co. Bulletins T7 and T9.
6. "Nuclear Energy as a Power Source", Rear Adm. H. G. Bowen, U.S.N.—"Mechanical Engineering", September 1946.

THREADING OF HIGH-TEMPERATURE BOLT STUDS.

By K. D. Williams*.

The threading of bolts, bolt studs, stud bolts, and nuts intended for use where temperatures of 850° F. and above are encountered has necessarily received much consideration by users and manufacturers.

Many activities, including the Navy, prepared standards for bolts, bolt studs, stud bolts and nuts for high temperature service, selecting the coarse thread series of threading for products up to and including one inch and the eight thread series above one inch. The fit of threads selected was the Class 3 fit as defined in the standards promulgated by the American Standards Association and the Interdepartmental Screw Thread Committee.

The selection of the Class 3 fit for both bolts and nuts insured assembly, without metal interference, but did not provide for a measureable clearance, in all cases, between mating threads.

No serious difficulties in the use of bolt studs and nuts, so threaded, were reported from the Naval service. This does not mean that seizing of threads did not occur, but simply that such conditions probably were not considered serious enough to report.

Seizing of assemblies threaded Class 3 were reported by many municipal power generating stations, employing steam temperatures of 850° F. and above, and also by the Petroleum Industry. Because of this fact, Subcommittee No. 7 on Screw Threads for High-Temperature Bolting of the American Standards Association, Sectional Committee B1, on Screw Threads (jointly sponsored by the American Society of Mechanical Engineers and the Society of Automotive Engineers) was appointed in November 1941. This Sectional Committee was charged with the responsibility for developing a proposed American Standard for the limiting dimensions of threads for this purpose which, as far as practicable, would eliminate the tendency to seize.

Subcommittee No. 7 developed a new standard for threading the bolt in which the bolt has been given Class 3 tolerances but

* Bureau of Ships.

with the minimum pitch diameter of the bolt equal to the minimum pitch diameter of a Class 2 bolt. Nuts threaded to Class 2 dimensions are provided to mate with bolts threaded the new Class 7 fit.

Because of lack of clearance between mating threads, where both the bolt studs and nuts are threaded Class 3, space is not provided for the retention of the anti-seizing compound during assembly. In threading bolts Class 7 fit and nuts Class 2 fit a clearance between the mating threads is provided in all cases. This clearance provides space for the retention of anti-seizing compound applied before assembly.

Calculations were made by the Bureau of Ships to determine just what effect the proposed dimensions of threads designated as Class 7 would have on the strength of the bolt studs. The results of these calculations indicated that the loss of strength involved would be insignificant.

To check the results of these calculations and determine if the proposed standards for threading could safely be used by the Bureau of Ships, tests were conducted at the U. S. Naval Engineering Experiment Station.

The following is quoted, in part, from the report of the tests:

"The Class 2 and Class 3 fit bolts and nuts were threaded by a manufacturer, whereas, the American Standards Association on Class 7 threads were prepared at the U. S. Naval Engineering Experiment Station. For the deformation tests, the bolts were threaded on one end only. In these tests the deformation in the bolt and nut threads actually engaged were determined. Full threaded studs with a nut at either end were tested in tension. The bolts were mounted in suitable fixtures so that the load was applied on the faces of the nuts. The nuts were positioned with one full thread of the stud extending beyond the top face of the nut.

"The four bolt and nut fit combinations investigated are presented in Table I. In each case, the thread dimensions for the maximum nut and minimum bolt were used. Each bolt and nut combination was loaded initially to assure proper seating of the nut and bolt threads. Deformation readings in the nut and bolt threads actually engaged were obtained. Tests were made using six, five, four and finally three threads engaged. This was accomplished by backing off the nut one thread after each set of

TABLE I

LOAD AND DEFORMATION DETERMINATIONS FOR SEVERAL BOLT AND NUT COMBINATIONS

(3/4" Bolt and Nut. 10 Threads per inch)

Case No.	Symbol on Graphs	Standard	Bolt or Nut	Class	Pitch Dia., in.		Clearance in.	Load to fracture, lb. Three threads engaged	Load to fracture, lb. # Six threads engaged
I	○	A.N.	Bolt	2	-	.6786	.0128	20800 ∅	24600 X
			Nut	2	.6914	-			
II	△	A.N.	Bolt	3	-	.6805	.0090	20500 ∅	24800 X
			Nut	3	.6895	-			
III	⊡	A.N.	Bolt	3	-	.6805	.0109	21000 ∅	25500 X
			Nut	2	.6914	-			
IV	X	A.S.A.	Bolt	-	-	.6786	.0128	20600 ∅	24750 X
			Nut	2	.6914	-			

A.N. -- American National
A.S.A. -- American Standards Association
∅ -- Stripped threads on bolt
-- Full threaded bolt
X -- Bolt fractured

3/4" BOLT AND NUT
10 THREADS PER INCH
LOOSEST CONDITION - MAX. NUT ON MIN. BOLT

SIX THREADS ENGAGED

CASE I -- ⊙ -- CLASS 2 FIT
" II -- △ -- " 3 FIT
" III -- □ -- " 3 BOLT - CLASS 2 NUT
" IV -- X -- A.S.A. BOLT & NUT

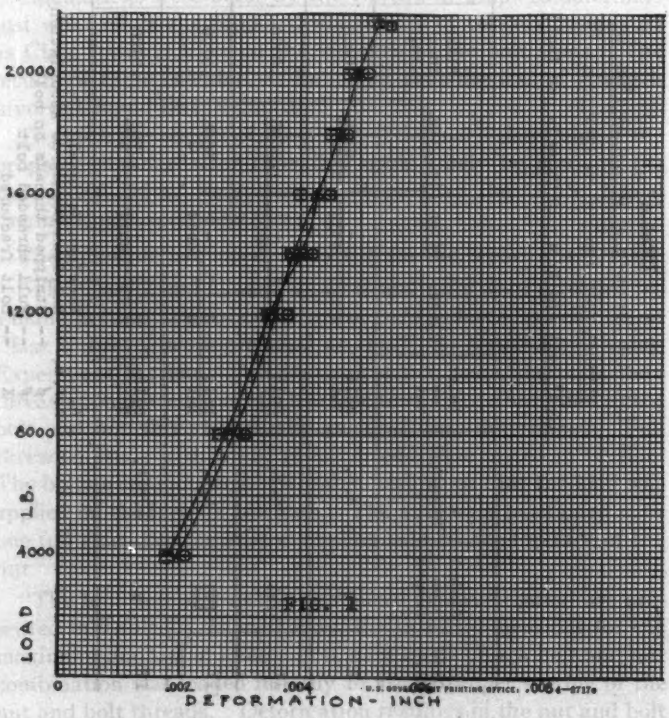


FIG. 2.

3/4" BOLT AND NUT
10 THREADS PER INCH
LOOSEST CONDITION - MAX. NUT ON MIN. BOLT

FIVE THREADS ENGAGED

CASE I -- ⊙ -- CLASS 2 FIT
" II -- △ -- " 3 FIT
" III -- ⊡ -- " 3 BOLT - CLASS 2 NUT
" IV -- × -- A.S.A. BOLT & NUT

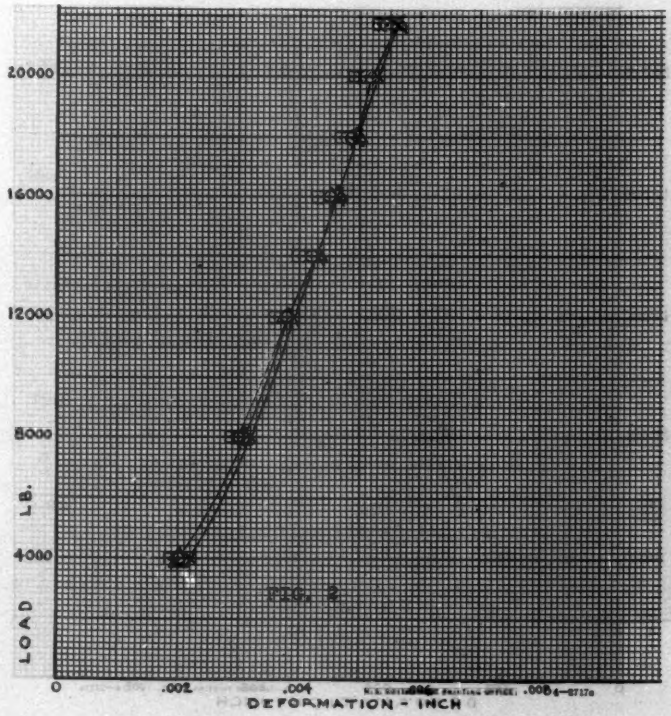


FIG. 2

3/4" BOLT AND NUT
10 THREADS PER INCH
LOOSEST CONDITION - MAX. NUT ON MIN. BOLT

FOUR THREADS ENGAGED

CASE I -- ⊙ -- CLASS 2 FIT
" II -- △ -- " 3 FIT
" III -- ⊡ -- " 3 BOLT - CLASS 2 NUT
" IV -- × -- A.S.A. BOLT & NUT

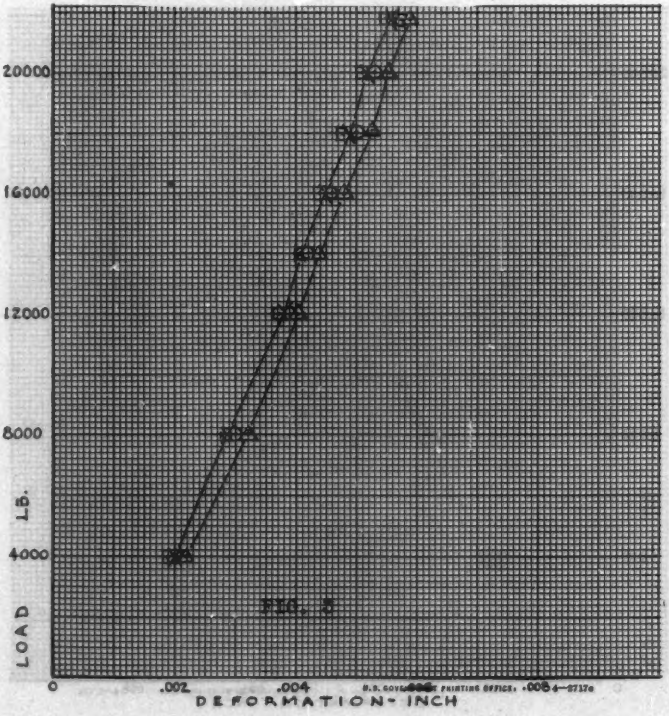


FIG. 5

U.S. GOVERNMENT PRINTING OFFICE. 16-27170

3/4" BOLT AND NUT
10 THREADS PER INCH
LOOSEST CONDITION - MAX. NUT ON MIN. BOLT

THREE THREADS ENGAGED

CASE I -- ⊙ -- CLASS 2 FIT
" II -- △ -- " 3 FIT
" III -- ⊡ -- " 3 BOLT - CLASS 2 NUT
" IV -- × -- A.S.A. BOLT & NUT

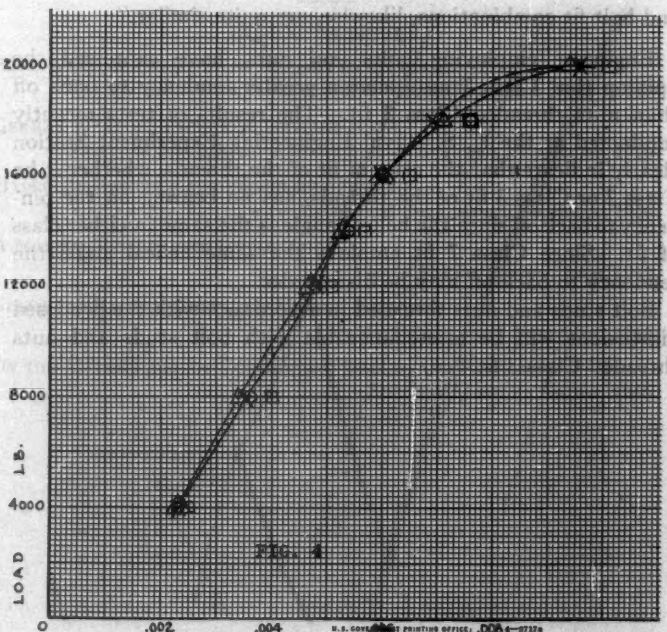


FIG. 4

deformation readings until only three threads were engaged. The deformation determinations are presented in Figures 1, 2, 3 and 4.

"The tests to determine the load carrying ability of a nut and bolt having only three threads engaged reveal that the tensile stresses necessary to strip the threads from the bolts for the American Standards Association and American National bolt and nut combinations are quite similar. The stresses necessary to fracture the full threaded bolts, likewise, show very little difference for the various nut and bolt fit combinations. A study of the Figures reveals that the deformation determinations for the American Standards Association and American National nut and bolt fit combinations, likewise, are quite similar."

Engineers employing bolts and nuts, have expressed the opinion that Class 2 nuts have a greater tendency to back off than nuts threaded Class 3 fit. The results of tests recently conducted at the U. S. Naval Engineering Experiment Station clearly indicate that the class of fit of the threads, whether it be Class 2 or Class 3 fit of the threads, has no bearing on the tendency to back off since the helical angle is the same in either class of fit. Since Class 7 fit employs the same helical angle the tendency to back off may be disregarded.

Bolt studs and nuts threaded in accordance with the proposed dimensions will be interchangeable with bolt studs and nuts threaded Class 3 fit.

NOTES.

INDEX TO NOTES.

SOME WARTIME EXAMPLES OF REPAIRS TO MERCHANT SHIPS.
—Transactions of the Institute of Marine Engineers, April 1,1946

GERMAN WARTIME TECHNICAL DEVELOPMENTS.
—Society of Naval Architects and Marine Engineers, November, 1946

HYDROGEN PEROXIDE FOR PROPULSIVE POWER.
—Mechanical Engineering, December, 1946.

A MARINE GAS TURBINE INSTALLATION.
—Pacific Marine Review, June, 1946.

STEAM JET REFRIGERATION FOR MARINE PURPOSES.
The Marine Engineer, November, 1946.

ON THE MATHEMATICS OF COMMITTEES, BOARDS, AND PANELS.
—Scientific Monthly, August, 1946.

SOME WARTIME EXAMPLES OF REPAIRS TO MERCHANT SHIPS.*

By C. Bartlett.

Synopsis.

The repairing of merchant vessels under the special conditions resulting from a world war presented entirely new problems to be faced by ports in many countries. This paper describes those problems and the way in which they were generally solved. This is illustrated mainly by quoting actual cases showing how damaged vessels were made seaworthy by temporary repairs carried out in odd places with limited facilities and at times by means of new methods and new ideas.

The lessons of this period of special emergency, in so far as ship maintenance is concerned, are also discussed and reference is made to the use of welding in ship construction as it affects ship repairing.

The General Problem.

Before the war ship repairing facilities throughout the world underwent changes from time to time, but these were usually of a gradual character and generally determinable in advance. They came about through alterations in trading routes or in the volume of trade on any particular route, while competition within the repairing industry itself sometimes caused slight upheavals, but the volume and distribution of work was not subject to violent and disruptive change. Machinery and materials were always available in the desired areas or were readily obtainable at short notice while the supply of skilled labor was equal to the task of keeping the machinery fully employed.

The 1939 war burst wide the whole of this peace time economy and distribution of the ship repairing industry. The regular trading routes for our ships became entirely altered and the Merchant Navy was scattered all over the seas. Stability gave place to fluid conditions liable to sudden and major modifications. Enemy submarines, mines and aircraft called for repair work of a major character in areas far removed from necessary facilities, and harbors and inlets all around the Continents became at least temporary refuges for badly crippled vessels.

As a consequence of this the need for ship repairing establishments arose in entirely new areas, while the call for these facilities in the regular repair areas fluctuated widely and rapidly. One of the chief difficulties resulting from these changes was that while ships could be diverted by a radio call, material and labor necessary for repairs were not so easily re-routed. For example, the almost sudden diversion of a large number of vessels when the Mediterranean was practically closed could not be accompanied by an equally quick diversion of the necessary repair workshops to ports around the southern shores of the Continent of Africa.

In such instances—and there were many of varying degrees of intensity—the men on the spot were called upon to do their utmost to keep the ships going, and great credit is due to the many small ship and engine repairing establishments dotted along the seashores of many countries for the noble efforts made in dealing with casualties both small and large. May it not detract from their share of credit if it is suggested that their work was facilitated and their efforts directed into the right channels by the existence of the peace time organization of Lloyd's Register of Shipping, whose skilled ship and engineer surveyors practically covered the world's seaboards. During the war these experts flew thousands of miles in answer to urgent calls for advice and assistance, and journeyed to all sorts of isolated spots where ships were in trouble. Sometimes the only help available on arrival was supplied by the ship's crews. Under such circumstances many vessels were saved which would have become broken backed wrecks in peace time. It is to be hoped that this wonderful work done by the officers and men of the Merchant Navy will one day be recorded by someone capable of performing the task.

* Reprinted from the "Transactions of the Institute of Marine Engineers"—April 1, 1946.

The determination, endurance, ingenuity, and loyalty to their ship, shown under disheartening conditions deserve the highest tributes.

The Effect of Enemy Action.

The loss of a merchant vessel in peace time is due, in the majority of cases, to an initial loss of buoyancy caused by the piercing of the outer protection of shell plating, deck plating, or of appliances covering essential openings in that protection. Normally this piercing is accidental, but in war the main object of the enemy's attack on shipping by submarine and surface ship, by mines of many types and by bombing, is to improve on the peace time methods of destroying the reserve of buoyancy. In addition his action sometimes resulted in the creation of so much main structural damage that the vessel became a total loss due to lack of strength.

At first the enemy was aided in his destructive efforts by the fact that merchant vessels are built to suit the requirements of peaceful trading and consequently were not the best that could be devised to combine their normal duties with maximum ability to withstand the effect of enemy action. Modifications were introduced at a very early stage in the war which gave increased protection to the vessel's buoyancy and stability. These alterations no doubt decreased the amount of ship repair work in so far as they limited the extent and possibly the frequency of damage due to enemy action. On the other hand their effectiveness certainly enabled many damaged ships to reach safety which would otherwise have been lost.

Deeper Loading.

The effective weight carrying capacity of the Allied Mercantile Marine was augmented, as a war emergency measure, by the introduction of deeper loading. The experience of the previous war suggested that this reduction in freeboard could be accepted as a war risk not entailing any abnormal hardship. Mention is made of this factor here only in order to add a few remarks about its effect on the volume of ship repairs.

In the first place it is to be remembered that the change was one which only increased the weight carrying capability of the vessel, and that in the average tramp steamer subject to the deeper loading this increase was of the order of five per cent. The number of times the nature of the cargo permitted a vessel to be loaded to her new mark is problematic and the evidence accumulated by Lloyd's Register did not include information on this important point. Therefore, any attempt at accurate analysis was impossible, but the records do not show that deeper loading led to any noticeable increase in the number of cases reported of structural damage due to heavy weather.

Various wartime precautions and regulations, including sailing in convoy, slinging out lifeboats in danger areas and making passages in Arctic waters under abnormal weather conditions produced an increase in the amount of damage to boats, ventilators, deck fittings and general equipment. It is likely that under such conditions deeper loading added its quota of risk, but such damage as has been mentioned cannot be included under the heading of structural damage.

Survey Work.

In normal times the main work carried out by ship and engine repair establishments was wear and tear repairs necessary at Special Surveys and general maintenance repairs carried out at that time and at the annual docking, and freeboard and boiler surveys. The Periodical Special Survey required by the Classification Society was a four-yearly event. To decrease the strain on the repair establishments in the early days of the war the Government decided that the thorough opening-up and examination of vessels at four-yearly intervals should cease. A modified form of Annual General Examination was instituted with the concurrence of the Classification Societies. In this examination those parts easily accessible were always examined, and of the parts requiring opening-up and cleaning before an examination was possible, only those likely to suffer from rapid deterioration were dealt with. The surveyors

recommendations as regards necessary wear and tear repairs in order to make the vessel efficient for a maximum period of one year were subject to license by the Admiralty officials. It can be placed on record that this arrangement was worked in friendly and amicable fashion by the classification surveyors and the Admiralty's representatives.

The difficulties met with in carrying out these General Examinations were mainly in connection with the survey of the bunkers. Here the natural call for an examination of this particularly vulnerable part of the ship's structure ran contrary to the economical requirement that vessels should, as far as practicable, fill their bunkers at ports abroad to ease the strain on British collieries.

Some of the older vessels did, as a matter of fact, give a certain amount of trouble which indicated that something more than the requirements of the General Examination was necessary, and in the later stages of the war the Admiralty agreed that in such cases a more extensive opening-up should be arranged to prevent constant delays for minor repairs in out of the way ports.

Repair Methods.

The efforts of the enemy to destroy reserve buoyancy in ships varied from the machine-gun bullet holes through the intermediate stages of started seams, and small damage due to near misses from the air to shattered sides and decks. Likewise, temporary repairs to the craft remaining afloat after an attack varied from the wood plug filling a bullet hole to the temporary wood patch equal in area to the side of a house, and from the welded patch to huge structural girders tying together what would otherwise have rapidly become separated and useless halves of vessels.

Reviewing the various means of repair utilized during the war leads to the conclusion that pride of place rests with a relative newcomer, namely welding. The main reasons for this are the portability of the plant required; the ease with which the work can be carried out under difficult conditions, and even underwater in urgent cases; the at least temporary efficiency of the work resulting, and the speed with which repairs can be effected. Many of these advantages will be illustrated later in what is probably the most effective manner, namely by citing actual cases and showing photographs and diagrams of the work.

Having given welding the honor due to its achievements, notice must be taken of what is probably the oldest friend of ships' captains whose craft have developed leaks—cement. Literally hundreds of vessels have made voyages, without the delays necessary to carry out proper repairs on the good old cement box. Many have run for months with cement covering leaks which might otherwise have proved serious.

One wartime innovation which deserves special mention as making an efficient temporary repair where other methods failed is the bolt-driving gun. The principal of this invention is probably well known. The writer has seen repairs which have been carried out by this means and can only praise the ingenuity shown. That a diver can fire steel bolts from a gun through half-inch shell plating to form a neat watertight job is undoubtedly a clever idea.

After this preliminary review it is considered that the best way to illustrate the type of repair work carried out all over the world during the war is to quote actual cases.

Special Repair Cases.

The difficulties of the world repair situation may be judged by the case of a 6700 tons gross tanker. This vessel was torpedoed forward of the bridge but managed to reach Colombo, 2000 miles away. A diver examined the vessel and found a large hole and extensive internal damage. Repair facilities at Colombo were unable to deal with even temporary repairs, especially as the Japanese were making rapid and almost unchecked advances at the time. The tanker situation was serious and the surveyor reported that the master and the crew were willing and anxious to sail the vessel to a more suitable port. The vessel made Mombasa, and there only temporary repairs could be carried

out to the longitudinal and transverse bulkheads. After these were completed the vessel sailed to Durban, where she was drydocked and temporarily stiffened for a voyage across the Atlantic to Mobile. There the necessary permanent repairs were duly carried out.

Another case bearing on the same point, but one which ranks high amongst major temporary repairs carried out during the war, both in regard to the character and extent of the work, was a 10,000 ton tanker which grounded off the coast of Skye. The damage forward was serious, but being a tanker she managed to reach the Clyde when refloated. An examination afloat showed that the bottom of the ship was practically torn off back to the No. 4 main cargo tank. This case is noteworthy in the first instance as a demonstration of the value of the compressed air system on tankers. The fore end was being carried on compressed air, and having seen the vessel it is easy to realize that but for this the fore end would in all probability have hinged about some point on the deck in way of No. 3 or No. 4 tank and the ship would have been lost.

The efficiency of this system in maintaining support for the fore end may be gauged from the following report from the surveyor. With the draught 21 feet 9 inches aft and 19 feet 6 inches forward and the fore end under compressed air one compressor was started up. The bows rose 9 inches in one hour. With the compressor shut off it took 24 hours for the draught to increase by one foot.

In the above condition the vessel went to the Bristol Channel and was placed in dry dock. The bottom shell and most of the framing of the bottom from the stem to the after end of No. 3 tank was missing right across the ship. The side shell and longitudinals on both sides were gone with it for heights varying from 7 feet 6 inches to 14 feet 6 inches from the base. With this had disappeared to the same extent the two longitudinal bulkheads and all the transverse bulkheads. After much discussion and calculation it was decided to effect temporary repairs to enable the vessel to cross the Atlantic for permanent repairs in America.

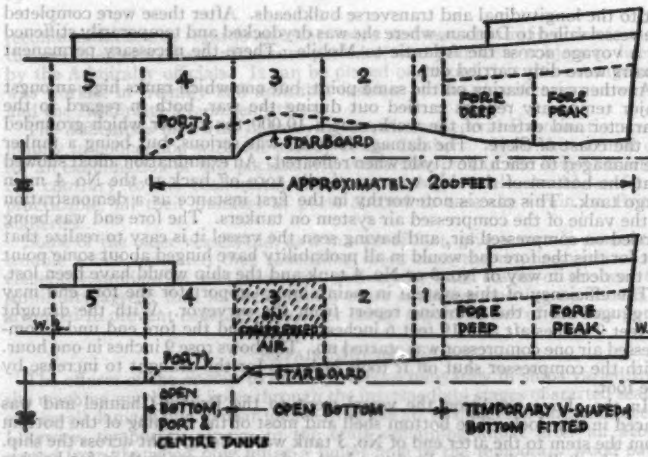
Buoyancy forward was necessary to help support the overhanging structure. To obtain this the fore peak, the fore deep and the pump room were fitted with a new V-shaped bottom and made tight. This buoyancy was then "attached" to the main hull more firmly and at the same time the ragged lower edges of the side shell and longitudinal bulkheads and the center girder were reinforced by heavy girders. No attempt was made to make Nos. 1, 2, 3 and 4 main cargo tanks tight. They were open to the sea. On completion of these repairs the compressed air system was once again called upon and the vessel reached Baltimore in safety.

It is impossible to give any details of the various calculations of relative strength trim and of distribution of ballast and compressed air in the vessel when she sailed, but Figure 22 shows her actual state when she arrived in the Bristol Channel and when she sailed for America. The calculations call for expert knowledge. Too many or too few compartments supported by compressed air might easily have had disastrous consequences in view of the hammer-head effect of the forward reserve buoyancy and the relatively low strength value of the temporarily reinforced structure.

Temporary repairs were not always the order of the day; in one case a major permanent repair was carried out by frabricating the new materials on the North East Coast and sending them piecemeal to Southampton, where they were erected in place, only closing plates to the shell and decks being worked on the spot. The vessel, a 5500 tons gross cargo steamer with machinery amidships, was beached in the English Channel after damage by enemy action. That part of the vessel forward of the after engine room bulkhead was salvaged and the new after end built on as indicated above.

During the war the Censor released some information regarding brilliant feats performed by masters and crews, and the example now quoted is worth recapitulation.

The *Hororata*, a vessel belonging to the New Zealand Shipping Co., was torpedoed about 220 miles off Flores Island in a rough sea. In a critical state she reached the Island only to find that the most elementary repairs were



(Above) Damaged condition on arrival in Bristol Channel.
(Below) Temporarily repaired condition on voyage from Bristol Channel to United States of America.
Figure 22.

impracticable. It was decided to risk the journey of 130 miles to Horta. There some cargo was discharged. Repair facilities were lacking but the crew cut down trees, made bolts and rivets, obtained some railway lines and fitted a reinforced wood patch over the hole. The cargo was reloaded and the vessel reached a British port in safety.

The report of the survey received by Lloyd's Register is interesting in its modest way. It reads—"We the undersigned Surveyors appointed by Dr. Antonio Zavier de Mesquita, Acting Lloyd's Agent at Horta, Faval, Azores, at the request of F. S. Hamilton, Master of the British Steamship *Hororata*, to hold a new survey on the above mentioned ship, to grant the referred steamer a certificate of seaworthiness do hereby declare and report as follows:—

"That we went on board the said steamer at 3 p.m. and after a final and careful examination of the repairs recommended previously, we found that the same had been carried out to our entire satisfaction, and the vessel is now in a seaworthy condition to proceed on her voyage to discharging port after which dry docking of the vessel will be necessary to carry out permanent repairs.

"That is all we have to declare on the present survey we held on board the said steamer which we sign at this town of Horta this 10th March, 1943."

(It is regretted that the signatures cannot be read.)

During a stage of the war when the enemy aircraft were paying particular attention to small coasters working their way up and down the East Coast, the writer happened to be in Hartlepool when one of these vessels that had been attacked arrived. This story is not of ship repairs but calls for no apology.

On going aboard it was found that the ship had been straddled by 50-pound bombs. Two had exploded in the sea on the port side amidships and one of these had torn a hole in the sheerstrake and bulwarks. Two dropped in the hold, which was empty, and two others fell just beyond the ship on the starboard side. The mate and one of the crew rushed down in the hold, picked up the bombs and carried them up the ladder. In the haste one of the bombs

was dropped on the deck but did not explode. They were both thrown over the side.

In view of the damage evident, the mate was asked if there were any more items requiring attention. He complained of a leak in the ceiling of his cabin but never mentioned his part in the previous escapade. The chief engineer was even more blasé. Asked how things were, he said that everything would be alright if it wasn't for the —— rheumatism in his shoulder.

Temporary repairs were obviously determined by the nature and extent of the damage, but in the majority of cases the work consisted first of emergency repairs to enable the vessel to be moved to a near port and then temporary work to send her to a repair port. Strength was usually made up by channels or lattice girders bridging the gaps in shell and decks, while bouyancy was obtained by fitting wood or steel patches. Emergency repairs were frequently made with material on the spot. Railway lines have already been mentioned, but hatch webs were generally found to be the handiest girder available for tying the decks. These were placed across the fracture, welded or bolted to the deck, and side support provided by brackets.

Except in extreme cases where no other means is possible it is advisable to keep the hatch web intact and fair the deck in way. If a piece has to be burnt out of the bearing flange to get the web to fay on the deck it usually fractures at such a spot. In one case the webs were fitted and rounded out over a small hump in the deck to move a vessel from a fairway where she was in grave danger of breaking in two. The operation was successful and the damaged ship brought to rest on a suitable bottom but the hatch webs only just lasted. They fractured where the pieces had been burnt out.

Illustrating the general method of temporary repair is the case of the 7100 ton vessel torpedoed in way of No. 3 hold during operations and beached after the damage. During salvage operations the bottom plating, except the keel, and the main and 'tween deck plating fractured in way of the holes in both sides of the vessel. To tie the vessel together the salvage officer fitted double channels to the main sheerstrake and the strake below on both sides of the vessel (Figure 1), while hatch beams were welded over the deck fractures on the main and 'tween decks to act as emergency girders (Figure 2). With this slender structural assistance the ship was brought into dry dock for more serviceable temporary repair to enable her to make the voyage from London to a North East Coast port.

To compensate for the fracture in the bottom shell, channels were welded to the outside after the break had been partially welded. The large hole on the starboard side was spanned on its lower edge near the bilge by a frabricated girder consisting of 14 inch channels 60 feet long, spaced 6 feet apart and tied together by vertical and diagonal struts and diamond plates. This girder was welded to the outside of the shell. It was also strutted to the tank top and from the after bulkhead of No. 3 hold (Figures 3 and 4).

The after bulkhead of No. 3 hold was too damaged to deal with, so a non-rule bunker bulkhead a few frame spaces abaft this was stiffened to withstand the water pressure, mainly by fitting a horizontal girder across it just above the level of the tunnel top. Other minor repairs were effected, including the bridging of the relatively small hole in the shell on the port side by a doubler. With the stiffening fitted by the salvage officer and work carried out in dry dock detailed above, the structural strength had been increased sufficiently to enable the vessel to proceed. As the vessel's trim and stability were satisfactory, it was not necessary to attempt to make the large hole reasonably tight. No 3 hold and the cross-bunker were left open to the sea.

When it became necessary to increase a vessel's buoyancy by making the damaged compartment approximately watertight, the wood patch came into frequent use. The general method used will be described in some detail in a later case, but Figure 5 shows repairs in progress on a 3600 tons gross tanker which was torpedoed by a submarine while running supplies into Tobruk in 1941.

The hole in the starboard side forward was 80 feet long and extended for practically the full depth of the ship's side. Tanks numbers 1 to 6 were open

to the sea. The vessel put back to Alexandria, was there patched up and sent to Suez. It was decided that the best use which could be made of the vessel, taking into account military requirements and the supply of material available, was to fit a wood patch over the hole after suitably framing it and use her as a water carrier.

The vessel, with patch, was seen by the writer in Gibraltar a year ago. She carried out her function of water supplier until the end of the war, the patch proving most efficient and presumably not being the cause of any special complaints regarding the brackishness of the water supplied to the troops in the desert. She is classed in Lloyd's Register as ✠ 100A1 "Carrying petroleum in bulk (but temporarily for water carrying service only in Western Mediterranean)", a notation suggesting that in a war emergency Lloyd's Register were as capable as any other organization of arranging a little patching in order to make its class truly descriptive.

Another wood patch case of temporary repairs was a "Fort" boat torpedoed near Cape Flattery while carrying a cargo of lumber. This is a Canadian job, and the dry docking was carried out with the cargo on board except, of course, that which had disappeared through the damaged side. The extent of the latter is well shown in Figure 6. The repair consisted of burning away the torn framing and plating, after which four 12 inch channels were fitted longitudinally to the shell and suitably placed in way of the hole which was between the bilge and the second deck. Three 12 inch vertical channels were also fitted and diagonally strutted to each other and to the tank top by 6 inch x 6 inch angles. The hole was then framed round the edges by plating to form a ledge to take the timber. To stiffen the planking suitably between the channels 10 inch x 10 inch baulks were fastened in place and the resulting framework was faced with 3 inch timber overlaid with canvas and then faced with 1 inch planking fitted diagonally. The patch was then caulked and is shown complete in Figure 7.

Other relatively minor damages were dealt with by welding or welded patches, the cargo was partially restowed and the vessel made the voyage from Esquimalt to Seattle. All the evidence shows that the Canadians made a neat and successful job.

Were this record confined to successful cases alone, the inference might be drawn that the surveyors recommending the repairs always played for safety. In a war emergency risks have to be taken and the rescue work on stricken vessels carried out without delay, otherwise it may be too late. Therefore the following case, in which the first attempt was defeated by bad weather, is added to balance the perspective.

A 12,000 ton vessel was struck by a mine while on a voyage from Fremantle to Adelaide. The damage was on the starboard side in way of No. 2 hold and caused the flooding of Nos. 1 and 2 holds. The vessel settled by the head and was abandoned. In the face of a considerable sea the crew returned and although seas were sweeping over the fore deck, a towline was made fast but parted. A second line was eventually made fast. Two lifeboats were lost during these operations, but the vessel made Port Lincoln drawing 45 feet forward and 12 feet aft, and with a bad list to port. A diver reported a hole about 50 feet long and 23 feet deep extending aft from No. 1 after bulkhead and up from the turn of the bilge. The shell plating was blown mainly inwards and the framing was missing. It is of interest to note how often the turn of the bilge limits the effects of an explosion.

Means for transhipping cargo were not available, but it was essential to reduce the draught forward by at least 10 feet. To accomplish this it was decided to cut off all projecting damaged material and fit a wood patch over the hole made by the mine in the shipside. A patch was built at Adelaide, partly dismantled, shipped to Port Lincoln, rebuilt on board and finally launched and secured in place by heavy wires and hook bolts. Divers fitted shaped timbers to the lower edge and the whole was caulked and covered by canvas. The water level was then reduced by 7 feet in Nos 1 and 2 holds Flooding brought the draft to 35 feet forward and 20 feet 3 inches aft.

The vessel left under tow, stern first. Unfortunately the pumps choked and

were unable to keep the water in the holds down. The patch thus lost the aid of the excess outside pressure and broke adrift when the weather became unfavorable. The vessel was forced to return to Port Lincoln. The removal of some cargo was then necessary, so that the ship could proceed to Port Adelaide. There an attempt was made to render the bulkhead watertight between Nos. 1 and 2 holds. This proved impossible, and it was decided to fit a steel patch over the hole. This particular patch was made of bolted channels fitted horizontally on the outside of the shell and bolted to stiffening inside.

Figures 8 and 9 show a type of outside steel patching which makes a very sound job, but requires proper facilities. This method was frequently employed when the work was carried out in a Royal Dockyard. In some cases the fact that the repair stood out beyond the ship's side or beyond the bilge made navigation alongside a quay wall rather difficult.

Another attempt fustrated by circumstances is worth quoting. A 9000 ton tanker was torpedoed in the North Atlantic in 1942 but managed to reach St. Johns, Newfoundland. A hole 60 feet long had been blown in the starboard side near amidships, and five strakes of main deck plating in way were badly damaged, in fact a large section of the deck was thrown up, landing further aft on the deck, demolishing deck cargo and pipe lines and fracturing the deck.

The vessel was loaded. Burning and welding apparatus could not be used, so bracing was bolted across the damage to enable the vessel to proceed to the nearest Canadian port. The intention had been to wait for favorable weather but the convoy date was fixed. After six hours at sea a heavy storm was encountered and the decks became coated with thick ice. The vessel returned to port, where an examination showed that some of the bracing had been carried away while further damage was found. The surveyor recommended that the vessel remain at the port until proper temporary repairs were effected, and the cargo be transferred to another vessel. Such a craft could not be obtained and as a matter of urgency a further attempt was made to reach a Canadian port after a diver had carried out an examination. The vessel broke in two and sank, but such a failure only serves to emphasize the bravery of the master and crew in attempting to deliver their cargo, even by means of a crippled ship, off the Newfoundland Coast in mid-winter.

After having seen hundreds of broken and battered vessels around the shores of the Mediterranean and under repair in this country, and having recorded two cases of comparative failure, this seems an opportune moment to state that one impression gained when examining temporary repairs was not that insufficient material had been added in the form of bracing, but that in some cases a tendency was shown to under-estimate the necessity for adequate attachment of the bracing to the main structure and especially to material removed from danger due to the possibility of cracks developing at raw and jagged edges. The rigidity of the shell or deck to which the bracing is attached is also of importance, while welding under difficult conditions and possibly with overworked, and therefore not fully efficient operatives, makes it desirable to have welded attachments of considerable length.

Taking the example of a hatch web welded across a deck fracture it is suggested that the total area of weld on the deck and of the same weld on the flanges of the hatch web should each be equal to at least 1.5 times the maximum sectional area of the web, only good and apparently sound welding being counted.

Returning to particular jobs, a 3900 tons gross cargo vessel was sunk by collision. The main damage was on the port side in way of No. 3 hatch, where the side plating from upper turn of bilge to the shelter deck was holed over a length of about 40 feet at the worst point. The main framing in way was torn or badly buckled; the upper and shelter deck plating and beams were cut into and torn away and No. 3 hatch coaming was buckled. On the starboard side the plating was set out over about four strakes and both decks in way were buckled.

The following temporary repairs were carried out. On the port side a ¾ inch temporary sheerstrake was fitted at shelter deck level to bridge the

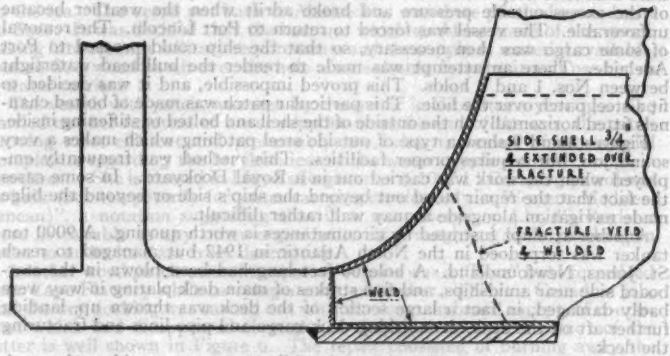


FIGURE 12.

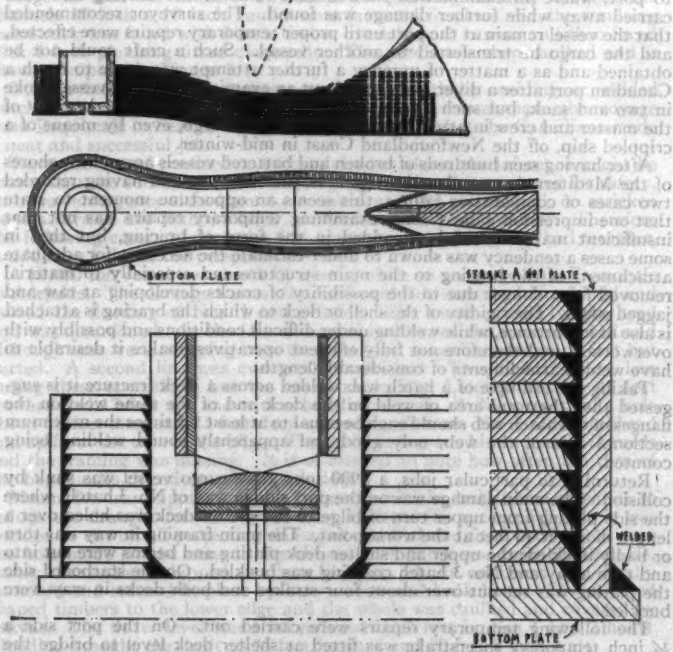


FIGURE 13.

gap in the shell. This was secured to the main structure by bolts but the butts were welded and the ends of this plating were welded to the existing plating. On this sheerstrake were fitted two 24 inch horizontal channels about 60 feet long. From upper turn of bilge to just below the deck the hole was covered by ⅜ inch plating, welded and supported by H bars and channels. The edge of the underwater part of the damaged side plating was flame cut and an attempt made to secure the bottom edge of the temporary side plating by underwater welding. This was prevented by the fast current and the muddy river. The attachment was finally made by shot bolts, using Cox's patent gun method. The vessel was then floated, the hold cleared of mud and water, and the plating welded and stiffened. The space between the sheerstrake and the ⅜ inch side plating was closed by wood sheathing, suitably supported.

The deck plating on both decks was stiffened by 24 inch joists, 65 feet long, welded to the decks and tied together. The gap in the shelter deck was closed by wood sheathing. The side shell starboard was reinforced by two 24 inch channels. Figure 10 shows 24 inch H bars fitted to the shelter deck. Figure 11 shows how the loss of continuity of strength is frequently compensated for, and calls for no comment.

The above cases give, it is hoped, a fair example of the means most frequently employed to deal in a temporary manner with extensive damage causing serious loss of structural strength or loss of buoyancy, or both. The work was planned and carried out under circumstances which made any attempt at calculation almost impracticable. Experience and common sense dictated the proper course to take, but even this was sometimes impossible to fully accomplish, owing to limitations in the materials available. Hence, to a very large extent, the differences shown in the ways of framing and strengthening.

A large number of serious bow damages occurred and frequently these were dealt with by the simple expedient of cutting away all the damaged material to a clean line, fabricating a V-shaped bow just sufficiently long to form an easy run for the water and welding this to the main structure.

Sternframes.

Not only were sternframes particularly vulnerable, but replacement was very difficult, consequently abnormal temporary repair methods were called for.

A *Fort* boat was dry docked in the Mediterranean with the foot of the sternframe set over 9 inches to port. The repairers built a coke fire around the bottom of the frame, got it thoroughly heated and pushed it back with jacks. The fire was then allowed to die out very gradually thus giving the stressed material a rough and ready form of heat treatment.

Fractures as indicated in Figure 12 were not uncommon. The acceptable method of treatment, provided the materials were available, is shown in the same sketch. One particular vessel so repaired gave no trouble between the time of the repair and of her sinking, eighteen months later.

One of the most elaborate efforts to fabricate a new sole piece to an open-type sternframe carrying a patent rudder was carried out at Suez. The new foot was actually built up from one inch plates welded together as shown in Figure 13. Only a war emergency could produce the necessity for such a repair. The missing part of the sole piece was 6 feet 4 inches long. The plates forming the replacement part were not only welded at the edges as shown, but also heavily slot-welded to each other. Admiralty "D" quality steel was used. On completion of the repairs a satisfactory steaming trial was carried out and the steering gear thoroughly tested.

At the other end of the scale, and this time from Newfoundland, comes the case of a small twin screw craft with a damaged sternframe which had to be replaced. The only material available having any semblance of being suitable was a crank web. The frame with gudgeons was marked out in spiral fashion on the slab of steel, flame-cut and then heated, uncurled and annealed.

Permanent Repairs.

Permanent repairs did not offer any particular scope in those areas where the pre-war repair facilities still functioned, but the development of repair works in new areas does call for some comment.

The first case concerns a new 2900 ton cargo steamer which grounded near Montreal, suffering extensive damage to side and bottom plating and framing in way. The vessel was hauled up on the slipway and then transferred to blocks. The damage was in way of the boiler room and Figure 15 shows the Canadian way of dealing with this problem. The boilers wre slid through the ship side, thus facilitating the work of renewal in way. Figure 14 shows the nature of the damage as a result of which 56 shell plates, 42 frames and floors, 29 tank top plates and 8 deck plates were renewed and a lot of incidental work carried out in 12 weeks.

From Canada to South Africa. Figure 16 shows a tanker torpedoed twice amidships. Figure 17 gives an idea of the damage above water. It extended down beyond the turn of the bilge. In this case a good deal of the work was fabricated ashore and lifted in place in long sections as shown in Figures 18 and 19.

The next picture, No. 20, shows the results of a collision between a cargo steamer and a cruiser. No comment is needed regarding the necessary repairs. It was practically a case of burning away all damaged steelwork and renewing the lot—in other words, the surveyor's delight.

Miscellaneous.

A vessel stripped her ahead turbine and was lying in a South African port unable to get the required replacements. The ahead turbine was blanked off and a left-handed propeller fitted so that the astern turbine could be used for going ahead. Trials were held, but the idea was not a success.

At a port in North Africa, repair facilities were meagre and the needed materials at a premium. The local surveyor formed a dump of plates and sections cut away during repairs or recovered from sunk ships. The repairer submitted his defect list, together with a statement of materials required. When this was approved he was permitted to rake over the dump. In this way, for example, a hawse pipe was temporarily renewed by a section cut from an old mast.

Figure 21 comes from Gdynia and shows how the Poles are improvising on account of the shortage of dry docks at that port. A sling is fitted around the ship and a 100-ton crane used to lift the stern sufficiently to enable the tail shaft to be drawn.

A serious shortage of anchors and cables caused many delays to vessels. While it is impossible to gauge the effect of various war emergency regulations on such an item as chain cables, it is certain that the loss of equipment during the war years was very high. With a return to more normal conditions it should be possible to eliminate some of the variants and thereby get nearer a solution as to the cause of the losses. It may be due to the reduction in the length of cable carried, to new types of shackles, to anchoring in strange harbors or other incidents. On such subjects as these Lloyd's Register keeps a careful analysis.

Welding has been mentioned in the beginning as the number one ship repairer's friend during the war. A great deal of experience has been gained, but the whole process is still developing and general conclusions are likely to become out-of-date in a very short time. Therefore only one point of interest will be dealt with. In the case of major damages to welded structures, the question often arises as to whether it is better to attempt to save some of the distorted material or cut it right away and prefabricate the whole. Experience suggests that the latter method results in a saving of time and produces better welding, but it is more costly in materials. Owners and classification surveyors would no doubt prefer the clean sweep method.

While on this subject, a true story may be quoted which draws attention to a simple device which has been brought forward before. A normal type of

FIGURE 1.

FIGURE 2.

FIGURE 3.

FIGURE 4.

FIGURE 5a.

FIGURE 5b.

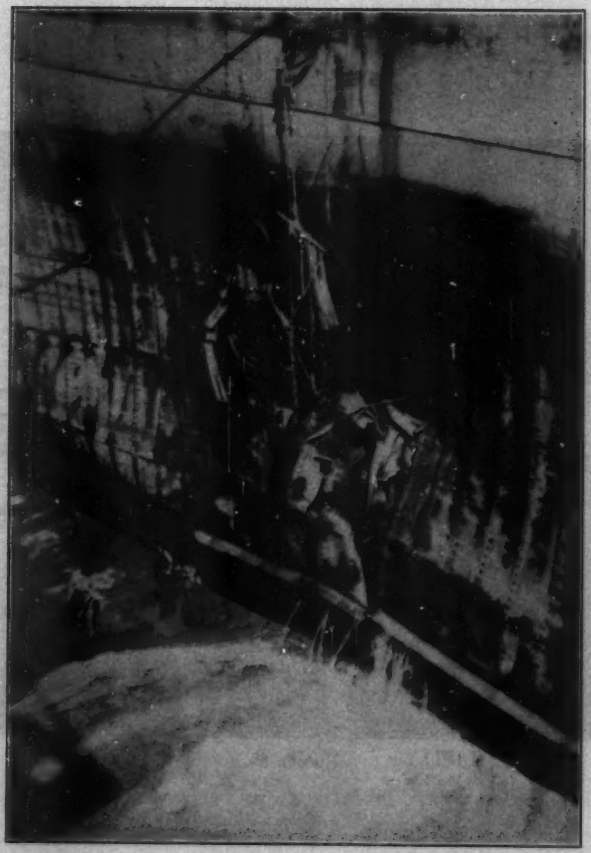


FIGURE 6.

FIGURE 7.

FIGURE 8.

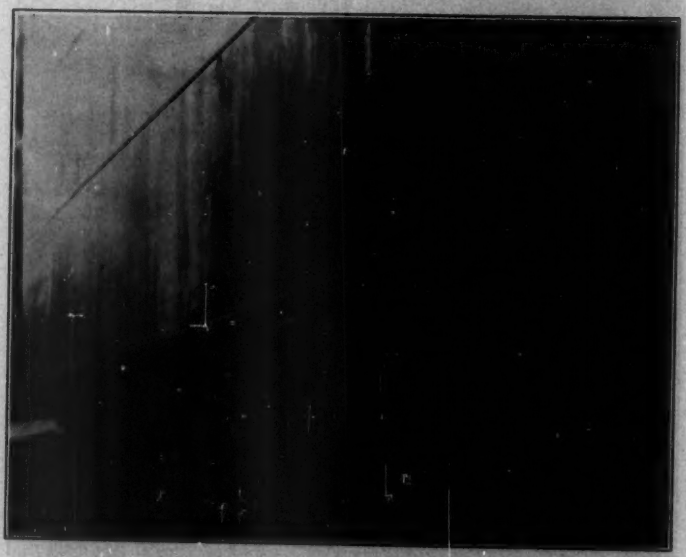


FIGURE 9.

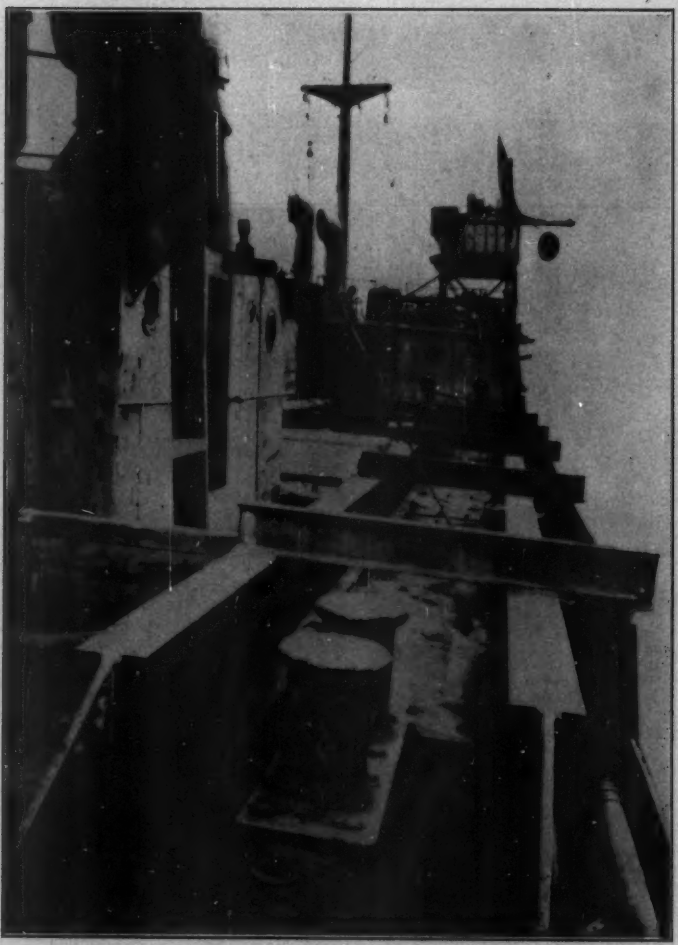


FIGURE 10.

FIGURE 11.

FIGURE 14.

FIGURE 15.

FIGURE 16.

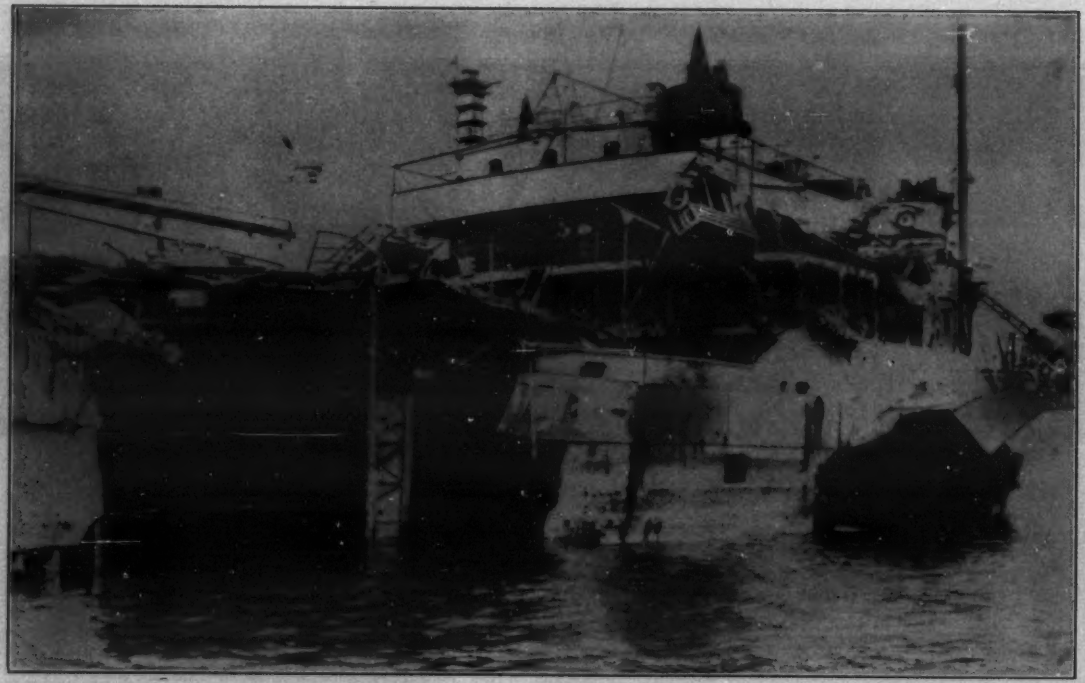


FIGURE 17.

FIGURE 18.

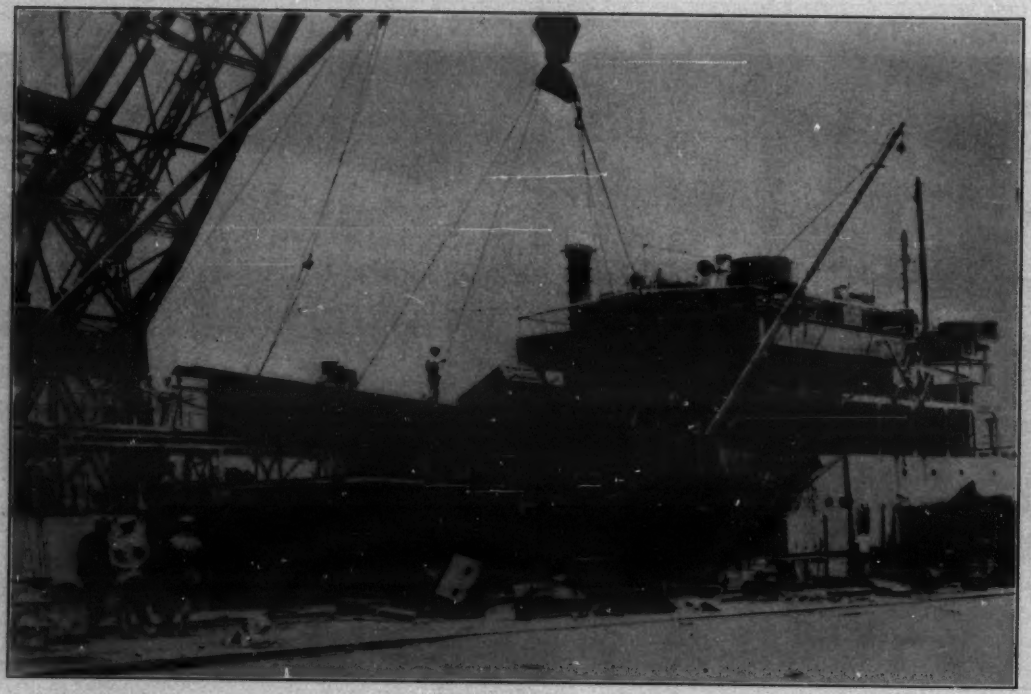


FIGURE 19.

FIGURE 20.

FIGURE 21.

welded cargo steamer with riveted frames arrived at a British port with the deck split from hatch corner across to the ship's side, the sheerstrake and part way through the strake below. One of the ship's officers said that it started as a small crack and they watched helplessly while it grew and grew. His attention was drawn to the fact that although it had gone so far and thereby greatly decreased the resistance of the structure to further extension it had actually stopped itself by running into a rivet hole. The full significance of the fact took a little time to sink in.

Conclusions.

Having studied various difficulties overcome during the course of the late conflict, the natural corollary is to consider how repair work might be facilitated should somewhat similar circumstances again arise. One major trouble was the shortage of dry docks. Floating docks are helpful because they are portable, but they are vulnerable to air attack. No doubt more particular attention will be paid to the number and the distribution of permanent dry docks in the future.

Repairs were sometimes held up by shortage of suitable material for use as girders to compensate for loss of strength in way of a damage. It should be a relative simple matter to devise a handy form of girder for use in such cases and ensure a supply wherever needed. This idea was put into operation in more than one sphere during the war and could surely be extended to temporary ship repairing. As has already been noted, hatch webs, heavy channels and joists were usually employed.

Compensation for lack of repair facilities is not easy, but why not floating repair workshops carrying their own flame cutting and welding plants, supply of useful materials and other necessities? Craft of this character of fairly high speed stationed at key points ready for instant service would, it is suggested, soon justify themselves.

Perhaps the most difficult temporary job was to make a large hole relatively watertight. Framing the hole and fitting reinforcements to support the patch could be standardized and sheathing designed for easy application. The chief trouble arises in proprely attaching the framing round the hole and in making it and the circumferential edges of the sheathing watertight. War experience shows the desirability of some such standardization.

The General Examination and its slight extension near the end of the war worked well, but had the war lasted much longer some further extension of the examinations required in older ships might have been considered necessary purely from the efficiency angle. Intermittent stoppages, especially at small ports, for minor repairs soon reduce the number of working days which a vessel can spend at sea. Therefore even from the economic angle some further tightening of the regulations might actually have increased the serviceable efficiency of such vessels, given, of course, the necessary man power.

GERMAN WARTIME TECHNICAL DEVELOPMENTS.

This article is a reprint of a portion of a paper presented by Commodore H. A. Schade, U.S.N., Director of the Naval Research Laboratory, at the Annual Meeting of the Society of Naval Architects and Marine Engineers on 14 November 1946. It contains considerable information on German naval construction and engineering which will be of interest to all naval engineers.

The United States Naval Technical Mission in Europe was organized for the purpose of investigating German science and technology for the benefit of the Navy Department's technical Bureaus and Offices. It was commissioned in January 1945 after preliminary organizational work had been completed. Its mission completed, it was decommissioned in November 1945. The principle of its operation was to provide an organization based on the continent of Europe which (a) would contain a nucleus of technical naval personnel which

could survey the field rapidly as the German collapse proceeded to determine what technical material of interest to the Navy deserved brief investigation; (b) could provide facilities to enable expert investigators in the various technical fields to conduct detailed investigations for the benefit of the Navy after the initial survey. The technical investigations were made both by technical officers of the Navy and by a large number of engineers and scientists employed by industry, educational institutions and other non-naval agencies. These individuals were under Mission control during their investigating period in Europe and their reports became the property of the U. S. Navy. A fundamental policy of the Mission required that all reports be written prior to the return of the investigator to the United States.

Because of the chaotic conditions on the continent at that time, the services which the Mission was required to provide for itself were extraordinary in their scope and variety. Air and ground transportation, weapons, special clothing and material, trucking, interpreters, translators and guides are but a few of the items of the complex system of services which had to be provided in order to obtain this information from the Germans.

The Mission was organized into two main branches—the Technical Branch and the Services Branch—the Services Branch provided the services outlined in the foregoing, and the Technical Branch consisted of the technicians who actually did the investigating work. The Technical Branch itself was divided into five sections. The titles are illustrative of their scope; namely, the Ship Section, the Ordnance Section, the Air Section, the Yards and Docks Section and the Electronics Section.

The Mission's main headquarters were established in Paris January 1945 and, as they became available later, field headquarters were established in Bad Schwalbach (near Wiesbaden), Bremen, Heidelberg, Munich and, finally, Berlin. Special air service was maintained to these points from the Paris headquarters so that no time would be lost unnecessarily in ground travel. The technical reports were written generally in Paris by investigators after their return from the field.

In this paper a description, general rather than detailed, is given of certain selected items which may interest members of this Society. These items form only a very small sector of the total field covered by the Mission, and, except for submarines, one of the less spectacular sectors. Matters of great interest to the Navy, although perhaps of less direct interest to this Society, such as aircraft design, electronics, guided missiles, V-weapons, fuel chemistry, supersonic aerodynamics, special ordnance and many other categories were explored and the results, in the form of reports, documents and samples, furnished the Navy Department. In particular, much extremely valuable material, still only in the research or development stage, was made available to the Navy through Mission investigations. These results of German resourceful and imaginative research will form a valuable stimulus to our own determination to keep our Navy technically alert and pre-eminent by means of progressive research and development programs.

The paper is, in general, a synthesis of information contained in Mission reports, principally those of the Ship Section. Since the author was Chief of the Mission, he prepared none of these reports himself and makes no claim to authorship or originality in their connection. Some of the sections of this paper were prepared in whole or in part by other members of the Mission and the author wishes to acknowledge help of this kind contributed by Captain L. V. Honsinger, U.S.N.; Captain C. S. Seabring, U.S.N.; Commander R. T. Simpson, U.S.N.; Commander L. A. Rupp, U.S.N., and Lieutenant L. H. Strauss, U.S.N.R.

A list of the pertinent reports together with information as to how further detailed data may be obtained is given in Appendix 1.

Incidentally, a considerable number of the investigators who worked with the Mission were members of this Society and to them the information given herein will be nothing new.

SUBMARINES.

General Design Features. At the beginning of the war, the Germans had a total of 57 submarines of three types—a coastal class, a medium-range operational class and a long-range operational class. Although no great submarine fleet had been built in preparation for the war, German designers nevertheless had gained more experience than the size of the fleet would indicate.

The Type VIIC was the first class of which none had been constructed prior to the war. A total of 572 vessels of this class were built, and it is interesting to note that 16 different shipbuilding companies participated in this program, only three of which built fewer than 20 vessels. Type VIIC was a vessel of approximately 700 tons surface displacement, 220 feet in length, with four bow and one stern torpedo tubes, a surface speed of 17 knots and a submerged speed of 7.6 knots. It involved no especially novel features.

This class was followed by the Type VIIC/41, which was similar to the previous class except that the designed depth was increased from 100 to 120 meters standard. (By German submarine builders' convention, the collapse depth of submarines was about 2.5 times the designed depth.) Eighty-seven of these boats were built, bringing the total number of VIIC's to 659.

A number of other classes followed, none of which was as numerous as those previously mentioned, until a turning point in German submarine design was reached in the summer of 1943. At this time, Allied counter measures were holding the German submarine fleet in check. The solution proposed was the design of a highly maneuverable, high-speed vessel which could remain submerged most of the time and which could operate at greater depths than previous conventional submarines. The problem was attacked with characteristic energy, and the result was a revolution in submarine design philosophy. The most interesting example of the new trend is the well-known Type XXI.

Leading submarine designers and engineers from all main building yards were conscripted in late 1943 and ordered to the little town of Blankenburg in the Harz mountains where the central design agency for the Type XXI program was established under the Speer ministry. All design work for this program was either done or approved by this agency.

Radical as these beginnings were, they did not prevent the Type XXI submarine from becoming a compromise with the more conventional approach. A prime reason for her construction was her high submerged speed, yet she also was given two 1700-horsepower engines with exhaust-driven superchargers in order to give almost equal surface speed; i.e., 16 knots as against 18 submerged. Later, when it was found that back pressure created in the Schnorchel kept the engines from obtaining their rated capacity, the superchargers were removed, since the Schnorchel was indispensable; and this reduced the engine output to about 1050 horsepower each. Advocates of the new design maintained that a true underwater boat had no need for a large tower and for anti-aircraft guns. While the tower is streamlined on Type XXI, it is still large; and there are hydraulically operated twin 30-millimeter anti-aircraft guns at each end of the bridge.

The vessel had a large battery capacity (372 cells, 11,300 ampere hours). The hull was designed for a depth of 120 meters, and one of these vessels had submerged to a depth of 190 meters under full control. Type XXI was the first German submarine to have six bow tubes with a capacity of 20 torpedoes.

Although a total of 119 of these boats were delivered by the assembly yards, none got out on war patrol. A persistent source of trouble was the hydraulic system, described later in this paper.

A still later type, the Type XXVI, was the ultimate in German submarine design, intended for continuous submerged operation. Type XXVI was to be a vessel of approximately 775 tons, 177 feet in length. Four separate means of propulsion were provided to drive the single propeller shaft; namely, Diesel engine, main motor, creeping motor (for ultra-quiet operation) and Walter turbine. The power ratings of these components are:

Turbine	7500 horsepower	Electric motor	580 horsepower
Diesel engine	1200 horsepower	Creeping motor	35 horsepower

These vessels were expected to make about 25 knots submerged on the turbine, and 10 on the electric motor; while for the Diesel power on the surface the speed was to be 14 knots.

A novel feature was its ten torpedo tubes, six of them amidships, bearing about 10 degrees abaft the beam. Serious thought was given to eliminating completely the conning tower, but instead she was given a fairly small streamlined one. Anti-aircraft guns were omitted. None of these boats was built. The first one was to have been completed in November 1945.

All types of German submarines had direct-driven propulsion rather than electric drive. In all earlier types, the Diesel engines and motors were clutched directly to the line shaft. In sharp contrast to this practice, the Germans undertook a new machinery design and arrangement on the Types XXI and XXIII submarines where reduction gears were used between the propeller and machinery. The main motors in both of these vessels have greater power than the Diesel engines. Several features were adopted aimed at a light main motor with a large power output. In Type XXI, by using a separate reduction gear, a high motor speed was utilized, thus giving a corresponding saving in weight and size. By increasing the battery voltage from the conventional 240 volts to 360 volts, a further saving was obtained.

In general, the new concept of a submarine as a vessel which could remain submerged for a long period of time did not involve radical changes in the design of the ventilation, air purification or oxygen renewal systems. The use of the Schnorchel at periodic intervals for charging batteries provided sufficient air renewal and ventilation. In fact, the total blower capacity in all German submarines, including the newest types, is relatively low; but the operational doctrine was such as to eliminate any demand for greater blower capacity. This should have affected the comfort of the crews, but apparently this condition was cheerfully accepted by them as necessary to permit safe operation with a minimum use of fuel.

The ventilation systems on all types of German submarines were fundamentally similar, consisting of one main supply blower, one main exhaust blower and one small recirculation blower. A noteworthy feature is that none of the blowers is placed directly in the supply or exhaust lines, but rather is connected across the lines so that all blowers are able to take suction from the main exhaust line and discharge to the main supply line. When the Schnorchel is being used, either main blower can take a suction from the air intake line on the Schnorchel mast and discharge into the air intake of the Diesel engines. When submerged, any of the blowers can be used for recirculation of air.

The operational doctrine was as follows. When the submarine first surfaced, both main blowers were run at full speed to ventilate the vessel. For normal surface operation, both blowers were secured. When operating on Schnorchel, the main exhaust blower was used while charging batteries; but, when not charging, neither blower normally was used. When submerged, not operating on Schnorchel, the blowers were used very sparingly; for silent operations, only the small recirculating blower was used.

In earlier submarines, no air conditioning was installed, and the large majority of operational submarines during the war had no means for cooling or drying the air in the vessel. In the Type XXI, an air cooling and dehumidifying system was provided which could be used also as a heating system.

In all German submarines, the battery ventilating system is connected to the ship's ventilation exhaust line without special battery blowers. The rate of battery ventilation and the periods during which it was done would be considered by us to be at a dangerously low minimum, although no serious battery explosions are known to have occurred. It is interesting to know that on all German submarines no smoking was allowed below decks either surfaced or submerged, at sea or in port. This rule was accepted cheerfully by the operating personnel as being an unquestioned necessity.

On the German submarines prior to Type XXI, hydraulic power was used only for hoisting and lowering the periscope, for training periscopes in cases

where fixed height-of-eye periscopes were installed, and for hoisting the folding-type Schnorchel mast.

On Type XXI, the Germans took an extremely radical step in extending the use of hydraulic power to include not only periscopes but the bow and stern planes, the rudder, torpedo tube shutters and doors, and the training of the deck guns. Several reasons have been given for the sudden change to the widened use of hydraulic power, viz:

(1) Electric motors were scarce.

(2) Parts of hydraulic systems could be made by various shipyards and plants, thus spreading the work rather than overloading one or two electric motor plants.

(3) A Dutch submarine had been captured which had made wide use of hydraulic power and served as a guide.

(4) The successful hydraulic periscope installations on previous types had "sold" the Germans on hydraulic power.

(5) A quiter installation than was possible with motors was expected to result.

(6) It was desired to take advantage of the greater efficiency offered by an overall hydraulic system over individual electric motors.

Whatever the reasons for reversing the previous practice may have been, the radical increase in the use of hydraulic power was ill-advised and was instrumental in slowing down the entire Type XXI submarine-building program. Aside from the effect of Allied bombing, the main reason given for the fact that not one of the 119 Type XXI submarines delivered between June 1944 and the end of the war ever went on a war patrol was "the hydraulic system was too complicated and clumsy."

In the earlier installations for periscope use, the power system consisted essentially of two pumps, the accumulators, a collecting tank and an automatic switch. The pumps were started automatically as the pressure on the hydraulic systems dropped to 44 kilograms per square centimeter and stopped when it reached 80 kilograms per square centimeter. The pumps took their suction from the collecting tank into which all oil from all parts of the system drained, so that, assuming an absolutely tight system throughout, the oil level in this tank would always be the same for a given state of discharge of the system. This fact was used in controlling the automatic switch for starting and stopping the pumps, which depended for its operation on the fixed relationship between the amount of oil in the collecting tank and the pressure on the high-pressure side of the system. Since this switch actually, therefore, operated on the small pressure differential due to the varying head of oil in the collecting tank, it had a weakness caused by the fact that it was too sensitive to rolling of the ship, particularly when the collecting tank had a large horizontal cross-section.

The hydraulic power plant designed for Type XXI was basically the same as that previously used, except that the pumps and accumulators had a greater capacity. The automatic switch, however, was modified so that its operation was based directly upon the pressure on the high-pressure side rather than the amount of oil in the low-pressure collecting tank.

The principal source of trouble in the hydraulic system in Type XXI was in the hydraulic motors and controls. For example, the hydraulic bowplane rigging and tilting mechanism originally was extremely complicated, including a temperature compensation element to counteract viscosity changes; and a hydraulic servo motor, so designed that when the operator put his operating level on the angle it was intended that the bow-planes assume, the system would come automatically to neutral when the planes reached that angle. These refinements were found finally to be impractical as well as unnecessary, and finally the system was modified to provide only for a direct system; i.e., the controller in the control room was positioned to feed high-pressure oil to one side or the other of the bow piston and the planes moved accordingly.

The other operating elements of the system went through a similar progressive simplification. Considerable difficulty was experienced with the entry of

salt water into the hydraulic system, particularly at the bow-plane rigging piston, at the gun positions and through the many hydraulic fittings through the pressure hull.

In general, it can be stated that the German designers who worked out the Type XXI hydraulic system apparently lacked design experience in this particular field. They gave insufficient weight to the necessity for rugged and simple construction required for submarine wartime service and utilized delicate and complicated mechanisms to obtain unnecessary refinements. The extensive use of external hydraulic piping connections, piping and pistons indicated a lack of appreciation of the leakage difficulties that could be expected.

Construction (*Type XXI Program*). An integral part of the enormous Type XXI program was the maximum possible amount of prefabrication, in widely dispersed plants. The vessels, being about 250 feet long, were larger than most previous types of German submarines, and the hulls were of all-welded construction. The prefabrication plan involved making eight main sections of the hull plus a small conning tower-bridge section. Weight, rather than dimensions, limited the size of each section, the heaviest of which weighed 165 tons when completed. There were three major steps in the building of the submarine and each was performed in a different type plant. The three types of plants were known as *section yards*, *fitting-out yards* and *assembly yards*. Thirty-three different plants were engaged in the program.

The structural steel hull work was assigned to boiler shops, bridge builders and other similar plants. Here also larger foundations and some hull fittings were installed. These plants, known as *section yards*, were selected so as to decentralize the work and make use of labor and capacity which could not otherwise share in shipbuilding. It was necessary that the plants selected be near water transportation because the German inland waterway system was the only practical means of moving the large sections. In many cases the plants selected for this work were unfamiliar with welding and, accordingly, the central design agency prepared very detailed fabricating, welding and erection instructions and sequences with strict tolerances of 2 millimeters (about $\frac{1}{12}$-inch) on the diameter of the strength hull for matching purposes with the adjoining sections of the hull. This tolerance was much too strict and was changed early in the program to 5 millimeters. Even so, in final assembly the periphery occasionally had to be opened or closed by torch and welding procedure. This tolerance was a source of considerable concern to all three types of yards because each could claim irresponsibility for poor fit. Accordingly, a written report on dimensions as shipped and received was required, and fitting and assembly plants employed a large number of liaison men to inspect and guard their interests. As a rule, each section of the design was assigned to at least four plants chosen with geographical consideration to the final assembly point. Because of the relatively inexperienced welders used, a very thorough X-ray examination of typical and all important strength welds was required. These examinations were recorded and passed along to succeeding fitting and assembly yards, and then to the Department files. Any faulty work subsequently found was cut out, rewelded and X-rayed as were any cuts required to provide matching of structural section joints.

The *fitting out plants* were generally shipyards which had had some experience in building earlier type submarines. The experience here was used to install all local wiring, auxiliary machinery, main motor and gear installations, steering and diving systems, piping, furniture, etc. The completeness of the work in this stage was amazing—even the final coat of paint was applied except at the butt ends which welded to adjacent sections. In a few sections, main bulkheads were located near the ends but in many such natural structural supports did not exist and temporary structural spiders were provided to aid in retaining the shape and to serve as a locus for the central alining hole. Many production line methods were applied during this phase of the construction. The structural sections were placed on carriages which moved periodically past fixed staging at which point the same operations were performed on each section. Jigs and special engine-installing dollies were devel-

oped and the advantages of access through the section ends, rather than through restricted hatches or soft patches, were appreciated obviously and utilized to the fullest extent.

The *assembly plants* were restricted to three first-class shipyards already experienced in submarine construction, plus a new plant which was never completed. They were Deschimag at Bremen, Blohm and Voss at Hamburg and Schichau at Danzig. Under normal construction methods, such ships would have been on the ways seven to nine months but by this process 60 days was generally the scheduled time. In this phase, the fitted-out sections were assembled to make complete submarines, and installations not previously practicable, batteries, main engines, long power cables, periscopes, guns, etc., were installed. Final testing was accomplished and the ships were turned over to one of several trial boards for thorough operation at sea before being turned over to the crew.

It was pointed out earlier that all transportation of the sections was by water. When a ship ways had been cleared by launching of a previous boat, the eight sections in order were brought around to the outer end of the ways and either slipped from a barge to the ways or lifted over by floating crane. Each section previously had had attached to its underside steel bracketed cradles with wooden slippers to match the ground ways, and incorporating lead screws and jacks which allowed adjustment in three dimensions and facilitated quick and easy alinement. Each section, from one to eight, was pulled up the ways by means of winches at the head end of the ways. It was customary to have the eight sections for another boat in approximate location on the ways eight hours after launching. The central alinement holes previously mentioned in bulkheads and spiders were now lined up by means of lights starting from section 5, and the seven circumferential joining welds made. These generally each took four welders one 8-hour shift, starting at quarter points and proceeding under careful control of a supervisor. These welds were subsequently X-rayed.

Thirty boats were to have been completed by July 1, 1944, and 233 by the end of the year. Actually none met the July 1 date and 80 were completed by the end of the year but were not in operation. It originally had been planned to produce 38 boats per month by December 1944, which was to have been the production figure from then on. Actually 28 boats were completed that month, but the over-optimistic first schedule plus Allied bombing interferences and the general military situation quickly reduced this maximum output to six in March 1945, the last month any were completed.

Some of the difficulties with this program have been mentioned. The plan work was much more involved than originally contemplated. In an effort to meet the unrealistic quotas, the section yards and the fitting yards would move assemblies along to the next yard in an uncompleted status which served only to magnify the troubles and eventually cause delays at the assembly yards waiting for the missing items. Because of difficulties of the inexperienced structural plants in producing sections within tolerance, the assembly yards tried to reshuffle sections to get a better match, and some confusion and inefficiency thereby resulted. The steel requirements were carefully planned at the beginning of the program, and it was not until many months later that delays were incurred because of failure to provide steel allocations for associated special structural hoisting rigs, cradles for the support of the sections, etc. Delays were encountered in the assembly yards by excessive repair work as a result of the turn of the war. Finally, the ships were more complicated and larger than previous boats, and a three-months training period for the crew was deemed necessary.

Figures 1, 2 and 3 show sections of Type XXI boats in various stages of fitting out and assembly.

An extraordinary part of this program which never reached completion was a completely bombproof assembly plant started in the summer of 1943 at Farga a few miles down the Weser River from Bremen. This reinforced concrete building was 1400 feet long, 285 feet wide and 59 feet clear inside. It was so large that it was difficult to find one vantage point to view the whole.

Although scheduled for completion in the fall of 1944, delays had set it back so that it was only about 80 per cent finished at the end of the war. Two bombs already had penetrated the 15-foot roof, where the second layer, bringing it up to 24 feet thick, had not yet been laid. All exterior walls were 24 feet thick and interior division walls were 6½ feet thick. The long dimension of the building was normal to the river, and bomb-proof doors at one side of this end opened to allow access from the river to a long wet basin (inner half, dry dock) in this corner of the building. "Fitted out" submarine sections were to be brought in by barge and unloaded from the wet basin to trucks running on four tracks. A turntable here and one on the other side plus lengthwise and athwartship track systems, two 200-ton cranes and many smaller ones permitted unloading the largest submarine sections for the Type XXI, weighing 165 tons, assembling all eight sections and moving the assembled boats through the shop to the completion stage. There was sufficient room within this building for 24 sections (3 boats) plus 13 assembled boats, each 250 feet long. Since the submarine-building schedule called for 30 days elapsed time within the plant, boats would be moved and one completed every 2½ days. To move from next to the final position (No. 12) on the rails to the completion stage (No. 13) in the dry dock, movable gates completed the watertight enclosure built in by walls around these two positions (Nos. 12 and 13), and the space was flooded until the submarine, in position 12, floated clear of its rail supports. Hauling the boat sideways to a position over the dock allowed de-watering to commence. The boats could then be either dry-docked or floated and, after final work, taken out through the wet basin through the opened bomb-proof doors and the short channel to the river. At no stage was there any exposure of either personnel or equipment. Over positions 10 and 11, two fully protected houses projecting above the roof housed 5-ton cranes and provided clearance for periscope installation.

A few miles from Farga, a huge underground storage area for submarine parts was built, apparently as a part of this overall project. At the end of the war, this consisted of about 20 finished bunkers and several in various stages of construction. These bunkers were of cylindrical steel riveted construction (arranged for oil storage eventually), about 200 feet in diameter and 40 feet high, and set with the tops a few feet below ground. The construction included a 3-foot concrete wall surrounding and a few feet away from the steel tank. These tanks were arranged with bins and shelves, and those seen were well filled with every conceivable type of submarine part. The bunkers were all interconnected at their floor levels with tunnels large enough to take medium-size trucks. Vents and manholes were provided in the top but were inconspicuous among the scrub pines and shrubs, scatter planted as camouflage overhead.

Surface Vessels.

Battleships. Although no modern German battleships were afloat at the end of the war, Mission investigators were able to find and study many plans of these vessels and to discuss with the German Navy Design Staff, remaining at Eckernförde at the end of the war in Europe, their thoughts and plans for the future ships.

The Germans recently had built two 42,000-ton battleships. The *Bismark* was built at Blohm and Voss, Hamburg, launched early in 1939, commissioned in September 1940 and sunk in May 1941 by units of the British fleet in the open sea. The *Tirpitz* was built to the same plans by the German Navy Yard at Wilhelmshaven, launched two months later than her sister, commissioned in February 1941 and sunk in Tromsoe Fiord, Norway, in November 1944 by bombs from an RAF attack.

These vessels were 821 feet long overall, had a beam of 118 feet and a draft of 35½ feet at 52,700 tons full-load displacement. They carried eight 15-inch guns in four twin turrets, two forward and two aft, and a divided secondary battery of twelve 5.9-inch guns for use against surface ships and sixteen 4.1-inch guns for primary defense against aircraft. There were, in addition, of course, automatic weapons of both 37 and 20-millimeter size. Two quadruple

FIGURE 1.—TYPE XXI SECTIONS.

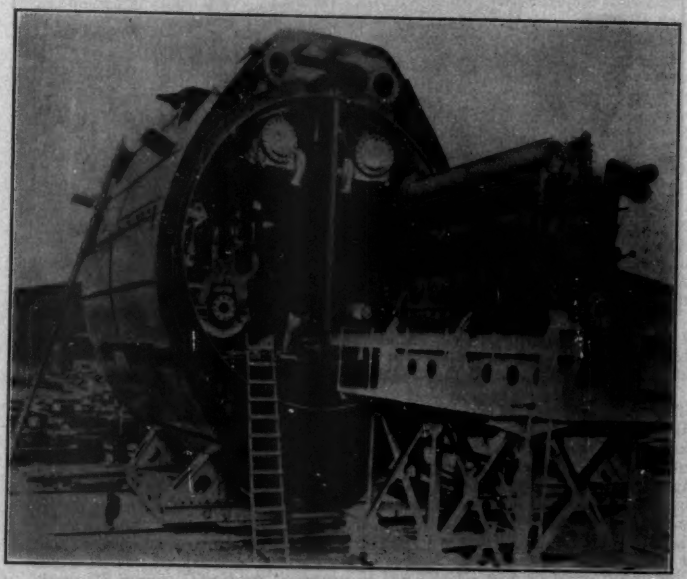


FIGURE 2.—TYPE XXI SECTION.

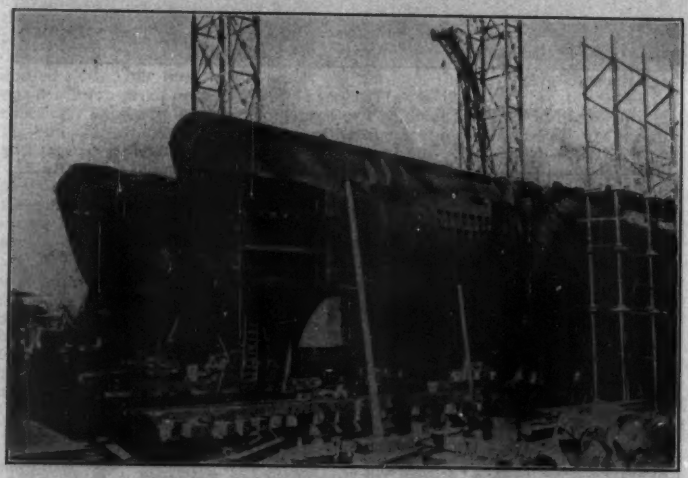


FIGURE 3.—TYPE XXI SECTIONS BEING ASSEMBLED.

torpedo tubes were provided on the main deck, and provisions were made to carry four scout planes.

The ships were triple-screw, with three sets of geared turbines totaling 165,000 maximum horsepower (150,000 horsepower normal rating), with a designed speed of 30 knots. Twelve Wagner boilers furnished steam at 850 pounds pressure and 840 degrees F. There were two athwartship arrangements of three boiler rooms each, with two boilers per space.

The hull, from clipper stem to cruiser stern, was virtually all-welded. Both main and second decks had sheer and 12 inches of camber. The main deck was 2 inches thick, all-welded, supported by transverse beams only. The second deck was a structural deck with both transverse and longitudinal beams, all-welded. The third deck was 3 inches thick, with sides sloping at 23 degrees outboard of the line of the inner torpedo bulkhead. On the slope the armor increased to about 4½ inches. The armored bulkheads were 7 to 8 inches thick; and the side belt, with the top at the main deck, was about 6 inches thick in way of the second deck spaces, 12 inches thick below this deck and tapered to about 7 inches at the line of connection of the sloping third deck to the shell. The inner torpedo bulkhead was carried to the main deck as a splinter bulkhead, 1½ inches thick. Three platform decks, all-welded, were provided forward and aft of the machinery spaces and an inner bottom located in way of the main machinery only, about 5½ feet above the keel.

The ship was divided into 22 watertight sections by 21 watertight or oil-tight bulkheads including the armored bulkheads. Transverse framing was spaced 1500 millimeters or about 5 feet, up to the side armor, behind which frames on half stations also were installed. The bottom had nine longitudinals on each side of the keel; the inner of two torpedo bulkheads joined the eighth, and the last one acted as a margin plate at the bilge which joined the inner bottom extended and the outer torpedo bulkhead.

The center shaft was supported at its after end by a centerline faired strut boss casting, riveted to the underside of the keel. The two side shaft ends, with propellers considerably forward to the center propellers, were carried by two-armed strut castings.

Twin balanced spade rudders had shafts inclined 8 degrees from the vertical, diverging downward. Rudders were supported by roller bearings. The lower edges of the rudders were approximately in the plane of the shafts and were about midway between the center and side shafts. The steering gear consisted of a left and right-hand spindle driven through a center worm shaft, controlled electrically by the Ward Leonard system. Each steering motor was capable of driving both rudders through a transverse shaft and coupling.

The ship was adorned with superstructures forward nine decks high. A relatively short but large single stack about mid-length of the ship appeared rather clear of superstructure and stowages. Combined boat and airplane cranes capable of a 12-ton lift were installed abreast the stack, port and starboard, over catapults capable of extending to 105 feet in length. A small hanger was built on each side of the stack and a larger one at the base of the main mast.

The ship was designed to carry a crew of 2100 officers and men, but, as seemed to be the rule in German ships, provision was made for a short trip for an additional 2500 men of whom 1600 could be slept in tiered portable bunks.

Contract plans were completed in July 1939 for a smaller type battleship or battle cruiser known as the "O" class, which was 32,000 tons standard displacement with a speed of 33.2 knots. Some material was fabricated, although no keels were laid. These ships had a length overall of 850 feet and beam of 99 feet. They were to carry six 15-inch guns, six 6-inch guns and eight 4-inch guns, all in twin-turrets, as well as smaller automatic weapons, torpedo tubes and four airplanes with provisions for four hangars. The main propulsion plant was to have consisted of a geared-turbine center shaft and two outboard shafts driven by Diesels, which by themselves were to drive the ship up to about 25 knots. The high economy of the Diesels was to have produced a large cruising radius. These ships were to have been 90 to 95 per cent all-

welded, including their struts, bossings and sternposts, which were to be fabricated weldments. They were to have had three rudders, with 31 feet spread from center to outboard rudder. Had this type ship been finished, it could have presented a formidable foe to surface escorts of our convoys to Eurpoe.

To provide the heavy backbone of the fleet, the German designers at this same time developed a design known as Battleship "H." These ships were expected to take 4 to 4½ years to build, and keels were to be laid in the summer of 1939. Blohm and Voss, Hamburg, was the lead yard and by October of that year, when work was stopped, had placed 1200 tons of steel on the ways, fabricated 3600 tons more in the shops and had another 12,000 tons on order. These ships were 910 feet long overall, 122 feet beam and at a full-load displacement of 55,440 tons were to draw 33 feet. The hull was divided lengthwise into 26 watertight sections. There were three rudders and four docking keels.

The machinery plants for these ships were to have been all Diesel. Four engines located at quarter points about a bull gear were connected to each of three shafts. The pinions were of flexible design developed by MAN, with spring torque transmission from pinion hub to pinion rim, acting as vibration dampeners. This plant had a rated output of 150,000 shaft horsepower (with the maximum 10 per cent higher) and at 260 propeller revolutions per minute was designed to drive the ship at 30 knots. At 19 knots, the radius of action was 16,000 miles. There were to be two oil-fired auxiliary boilers below and two exhaust-gas boilers above the armored deck.

These ships were designed to carry eight 16-inch guns, twelve 5.9-inch guns and sixteen 4.1-inch guns, all in twin-turrets. The main side belt was 12 inches thick.

Although work on the above ships was stopped in the fall of 1939, continued thought on this type produced in 1940 a scheme to salvage the already fabricated inner bottoms; and by adding 5000 tons to the design, 2000 tons of which was to have been in increased armor deck weight, produce a better protected vessel with 1 knot loss in speed. As the war was then going so well that ships for fighting several years hence seemed unnecessary, no action appears to have been taken on this design.

The following year, the designers again proposed a revision of the original plans known as H41. This ship would use little of the original "H," as its inner bottom was triple, 9.8 feet from the outer hull. Blohm and Voss undertook model experiments to determine methods of welding this triple inner bottom, which were reported to be successful, but no construction was started. This ship's principal dimensions were 902 by 128 by 59 feet with a draft of 39.9 feet, corresponding to a full-load displacement of 76,000 tons. By overloading the turbines, it was expected that 30 knots would be reached on 195,000 shaft horsepower. Again there were only three shafts; the center, turbine-driven, at 55,000 shaft horsepower standard (75,000 overloaded), while the wing shafts each would be driven by four 15,000-horsepower Diesels. The outer (Diesel) shafts were expected to drive the ship at 22 knots and produce a 20,000-mile cruising radius at 19 knots. The deck armor was 7.9 inches and the side belt 12 inches. The ship had a primary battery of eight 16-inch guns.

In 1942, design H-42 reached 1000 feet between perpendiculars, 141 feet beam and a full-load displacement of 98,000 tons. A speed of 32.2 knots was expected on 280,000 shaft horsepower, using four shafts for the first time; the outboard Diesel-driven shafts as before, and two inboard geared-turbine shafts of 80,000 shaft horsepower each. This design had a 13-inch armored deck and a 15-inch side belt. Its guns were still 16-inch, but it was to have carried six planes.

In 1943, this design was modified to a length, beam and draft of 1082, 158 and 42.2 feet, corresponding to 120,000 tons. The power plant duplicated the previous design and speed dropped off to 31 knots. This ship, known as H-43, was to have carried eight 19.7-inch guns.

In 1944, preliminary designs were prepared for the ultimate warship—1132 feet long, 169 feet beam and 69.2 feet deep with a draft of 44.3 feet. With

the same propulsion plant as in H-42, the speed was to be 30.1 knots at 141,500 tons.

The growth of the foregoing ships in their design characteristics appears to be a reflection of the hysteria by which the German leaders must have been gripped increasingly as the tides of war receded.

Prinz Eugen. The *Prinz Eugen* was the only large combatant vessel of the German Navy in good operating condition at the end of the war. One of five sister ships (*Blücher, Seydlitz, Admiral Hipper* and *Lützow*), she was launched in 1938 and completed in 1940. With the battleships *Scharnhorst* and *Gneisneu*, she made the famous dash from Brest through the English Channel in February of 1942, reaching Norway undamaged.

This ship was allotted to the United States and possibly was seen by many members of this Society in United States waters earlier this year. She displaces 19,550 tons fully loaded (tanks 85 per cent) and has an overall length of 692 feet, a beam of 71.2 feet and a draft of 24.8 feet. Her speed was rated as 32.5 knots with 133,630 horsepower (designed 132,000). She carries eight 8-inch guns in four twin-turrets, twelve 4.1-inch guns in six twin-mounts, four sets of triple 21-inch torpedo tubes and many smaller caliber AA guns.

The shell of this vessel is all-welded with the exception of one riveted seam near the neutral axis. She carries a main armor belt of 3.15 inches and has a 50-pound armor deck. The ship has a bulbous bow and an unusual bilge blister construction fairing into a rather open stern with three propellers and a single rudder.

The ship is fitted with anti-rolling tanks. Gyroscopic control equipment made possible a shift of 240 tons of fuel or water from wing tanks on one side to the opposite side by means of a high-volume air-blower driven by a turbine. The gyroscopic control acts on an oil pressure system with a servo motor which controls the air duct flap, which can be operated also by hand. In case of a list, and if so set, the governor automatically induces a shift of ballast water to the opposite side. During gunnery practice an artificial roll of 8 degrees could be obtained.

The fire main is copper with steel wasters. It was reported that it had required virtually no repair since the ship was commissioned. Piping for cooling, sanitary and warm washing water is also made of copper. Piping for flooding, drinking water, cold washing water and draining is galvanized steel. Remote control valves in general appeared to be operable entirely by means of rods and universal joints. No flexible shafting was in evidence, nor were there any remote hydraulically operated valves.

The ship's gasoline system consists of four cylindrical tanks of about 1200 gallons each, located two on each side in bilge compartments amidships. The piping from these tanks is double-walled with inert gas (nitrogen) at 3½ atmospheres between the two walls. When the pressure drops below 1½ atmospheres, a pressure switch stops the gasoline pump and a suction pump automatically drains the line, discharging into the gasoline storage tank. The tanks are of stainless steel. Nitrogen is used to cover the gasoline in the tanks, but at a pressure of only a few inches of water.

Crew's berths were installed for 1050 men and, in addition, a special type of portable berth is carried for 900 men. The latter was fabricated of pipe and canvas and capable of disjointing for storage. When needed, they are hung in tiers of three on cables with turnbuckles stretched between deck and overhead. The idea lends itself to the ready availability of additional berths so often needed on naval ships.

Magazines are interesting in comparison with those of our design. For example, wood chafing was generally applied to all shell bins. The sprinkling systems are relatively simple in that they extended over only a few parts of the magazine, and the holes and distribution of water appeared to have been left to the mechanic on the job. A float-type cut-out valve was installed to stop the sprinkling when the water neared the top of the compartment. Auxiliary lighting consisted of candles in glass-enclosed lanterns. Flame-proof scuttles are non-existent, although a guillotine-type of quick-closing passing port is installed between the magazine and handling room.

The main engineering plant consists of twelve Lamont boilers fitted with air preheaters and located in three boiler rooms. Fuel, air and feedwater are controlled automatically by a system designed and constructed by Askania. Although the boilers were designed for 1012 pounds, the normal steaming pressure was 750 pounds with 840 degrees F. total temperature. The boilers were equipped with two Saache burners, which were considered highly satisfactory by the Germans and have much to recommend them in the way of ruggedness and simplicity of operation. Each boiler room has a centrally located Askania control board with one panel for each of the four boilers in that room. Each boiler could be controlled independently from its own panel or all boilers could be controlled from one panel. If the load were reduced manually on one boiler, the load on the others was increased automatically at the same time.

The main engines consist of three geared-turbine sets, designed for 44,000 horsepower each. Each engine has one high-pressure, one intermediate and one low-pressure turbine; the intermediate-pressure and low-pressure turbines also house an astern turbine which develops 15,000 horsepower for approximately 30 minutes. The propeller revolutions per minute at full power are 275, and the turbine speed was reduced 12 to 1 through a single-reduction gear, connected to the line shaft through a clutch. The machinery spaces are crowded and accessibility of the units is relatively poor. An outstanding feature is the extensive instrumentation and the use of relatively small rectangular gages which take little space, make a neat board and are easy to read because of careful arrangement.

The electrical installation consists of five turbo-generators and five Diesel generators, one each of which is alternating current. The ship's light and power is 230-volt direct current. There is no casualty power supply system. In case of damage to normal leads, power could be supplied by stringing portable electric cables direct from the switchboard through armored ducts.

S-boats. The fast small torpedo boat in use in the German Navy at the end of the war, known as an S-boat or Schnell Boot, was the culmination of design studies and experimental development covering the preceding ten years. The outstanding design agency was Friedrich Lursen & Company, of Bremen Vegesack. Both two and four-cycle Diesel engine installations were tried, MAN's 11-cylinder, double-acting, two-cycle type being discarded in favor of a Daimler-Benz 20-cylinder four-cycle unit.

The latest design to have proved itself in service, the S38 type, was 118 feet length overall, 112 feet between perpendiculars, had a 15.7-foot beam at the waterline and a standard displacement of 104 tons. The draft at the bottom of the keel was 4.9 feet, and the maximum draft at the lower tip of the propeller 5.6 feet. Powered by three Daimler-Benz 20-cylinder Vee-type 4-cycle Diesels of 2500 horsepower each, speeds up to 43 knots were attained. With experimental engines of the same general type but of improved design, delivering 3000 horsepower each, speeds of 45 knots were attained on test.

The boats were not built for planing but had conventional lines with a forecastle deck. They were divided into eight watertight compartments as follows: (1) after peak and after fuel tank, (2) crew's quarters and galley, (3) middle fuel tank room, (4) after engine room with one engine installed, (5) forward engine room with two engines, (6) forward fuel tank room, wireless room and officers' cabin, (7) petty officers' quarters, (8) fore peak.

The hull was of double wooden shell construction, the inner shell being of white cedar of 12 millimeters thickness, the outer of copper-bolted mahogany fastened to bent wooden transverse and longitudinal frames. A layer of muslin was inserted between them. The sheer strake as well as the edge of the forecastle at the openings of the torpedo tubes was of solid oak. Light aluminum alloy was used for inside frames, bulkheads, superstructure and web frames in the engine rooms. The engine foundations were steel weldments. The forward collision bulkhead was of galvanized steel. For partitions, doors and furniture, waterproof glued plywood with inserted balsa wood core was used. The balsa wood core was protected against crushing by inserting strips of

harder wood. The result was a costly method of effecting a slight saving in weight.

The struts were of cast steel, all three shafts having main and intermediate struts. The main strut of the center shaft was arranged between the rudder and propeller and formed support for the rudder.

The armament consisted of two fixed torpedo tubes and a variety of small guns. The after deck was constructed strong enough to support firing equipment for depth charges on each side and a mine-throwing device. These were installed only for special missions.

One main and two side rudders were installed, the side rudders being termed effect rudders or backing elements. By means of a worm drive, actuated by a handwheel, the side rudders could be swung out during service to an angle of 30 degrees and made to act as backing elements. When swung out at high speed, the aerofoil-shaped rudders caused a sudden flow-breaking to take place inside the wake, resulting in the formation of an air-filled hollow space, with resultant influence on the direction and acceleration of the water as well as the stern trimming. This phenomenon, apparent only at high speed, was based on the flow structure in the vicinity of the propellers and behind them, and resulted in the following three effects:

(1) The revolutions per minute were lowered, or the power increased at constant revolutions per minute, as a result of the retained wake. The propeller efficiency was improved and the speed increased by about 1 knot.

(2) The high stern wave, lifted about 10 meters behind the ship, was flattened and other wave formation decreased.

(3) The stern trim of about 2½ degrees at maximum speed was reduced to practically zero.

Flow-breaking usually did not occur below 25 knots, but when it did occur it formed suddenly and could be determined easily by the operator. Having achieved an efficient flow cut-off and a large hollow space, both rudder angles could be decreased to about 17 degrees. At speeds below 20 knots, the effect had to be observed closely, as it often disappeared completely. Backing elements then had to be swung out again to 30 degrees and then reduced to 17 degrees; otherwise the desired effects were not re-established at higher speeds.

Turbines and Gears.

In view of the apparently low priority which German authorities assigned to their "big ship" Navy during the war, it is not too surprising that our investigators should have found their turbine propulsion equipment uninspiring and in most respects inferior. As compared with American machinery, the following assertions seem well substantiated:

(1) Overall fuel consumption rates are higher.
(2) Turbine heat and steam rates are higher.
(3) Reduction gear efficiencies are poorer.
(4) Weights per unit of output are greater.
(5) Space requirements per unit of output are greater.
(6) Accessibility for inspection, maintenance and repair is far poorer.
(7) Reliability is probably poorer.
(8) Difficulty of production is greater.
(9) Difficulty of large-scale reproduction is definitely greater.

All manufacturers used single-reduction gears with at least three (and sometimes four) pinions meshing with the main gear. The low turbine speeds resulting from this limitation on gear ratio contributed importantly to the high specific weights encountered, as also did the large number of turbine cylinders used in order to get more pinions to bear on the main gear.

Steam conditions were chosen (840 degrees F. temperature with pressures ranging from 700 to 1800 pounds per square inch gage) which, if properly utilized, should have produced high propulsion efficiencies. However, because of relatively poor turbine performance (attributable both to low speed and mediocre design and manufacture), higher than necessary gear losses and a

poor utilization of energy in the heat cycle (partially due to low efficiency auxiliary turbines), the fuel rate for propulsion only, without ship's service, was in the neighborhood of 0.7 pound per shaft horsepower per hour at full load. When lack of alloying materials for boiler superheaters forced the reduction of the initial temperature to 740 degrees F., this fuel rate became even worse.

A few examples of specific plants will illustrate these generalities. In accordance with the general practice of Blohm and Voss in Hamburg, the turbine design of the *Bismark* followed Parsons' principles rather closely. Each shaft, of which there were three, was rated at 50,000 horsepower normal, 55,000 horsepower maximum. The turbine arrangement was one frequently used for the larger vessels; three cylinders with the high-pressure and intermediate-pressure located aft of the single-reduction gear driving side pinions, and the double-flow low-pressure ahead of the gear driving the center or top pinion. Both the high-pressure and the intermediate-pressure had two-row Curtis first stages, but the rest of the steam path was reaction. The backing turbine consisted of a two-row Curtis high-pressure element in a separate casing located aft of the reduction gear driving the center pinion, and exhausting to a four-row reaction element in the after end of the low-pressure ahead casing.

The *Prins Eugen* had its turbines designed and built by Krupp Germaniawerft. The arrangement used in this case was most interesting and included many novel features. The gear is the orthodox three-pinion single-reduction, with the high-pressure and intermediate-pressure elements mounted aft and driving the side pinions and the low-pressure forward driving the center or top pinion. The backing turbine consists of a two-row Curtis high-pressure element in the forward end of the intermediate-pressure casing which exhausts to two single-row low-pressure elements, one in each end of the low-pressure casing. The high-pressure ahead turbine has a two-row wheel in the center supplied by four groups of nozzles, two for cruising and two for intermediate powers. Each of these groups of nozzles had its separate control valve. At cruising powers, the steam leaving the first stage goes through two single-row (Rateau) stages on the after end of the machine and is then brought around to three more Rateau stages on the forward end of the machine, exhausting at the extreme forward end of the casing to a double-flow intermediate-pressure and thence to a double-flow low-pressure turbine. At high powers, the two groups of single-row stages in the ends of the high-pressure turbine are put in parallel instead of in series and the exhaust from each end is combined at the inlet of the intermediate-pressure turbine. The changeover from cruising to high power is accomplished through the use of a complicated system of three interlocked valves with what appears to be an excessive amount of connecting piping.

The intermediate-pressure and low-pressure turbines are both double-flow reaction machines, each having a series of trick radial guide vanes said to reduce the number of stages required by at least two for each turbine.

The destroyer-building program is an interesting study from the standpoint of propulsion machinery design. This program started in 1935 with the Z1-4 built by Deschimag. These were twin-screw ships with machinery designed for 30,000 horsepower normal and 35,000 horsepower maximum per shaft. Steam at 1050 pounds per square inch gage and 840 degrees F. total temperature was supplied by Wagner boilers using Saache wide-range burners. The gear was a three-pinion single-reduction gear having idlers between the bull gear and the side pinions to give greater separation. There were four casings all forward of the gear: a seven-stage cruising turbine connected by a single-reduction gear and a Vulcan clutch to the forward end of a four-stage high-pressure turbine which drove one of the side pinions; a seven-stage intermediate-pressure turbine driving the other side pinion; and a low-pressure turbine having five single-flow and two double-flow stages driving the center or top pinion. The astern element consisted only of two single-row single-flow stages in the after end of the low-pressure casings. All stages were essentially impulse

with small amounts of reaction in the latter stages of the intermediate-pressure and throughout the low-pressure turbine.

The next class of destroyer, the Z5-8, had essentially the same power plant as the Z1-4 except for the omission of the cruising turbine, since on Z1-4 it was found that the windage in the Vulcan coupling, even when completely drained, was sufficient to over-speed the cruising turbine.

Concurrent with the building of Z5-8 at Deschimag was the construction of the Z9-16 at Blohm and Voss. These ships were equipped with the spectacular 1800 pounds per square inch gage Benson boilers and considerable thought was put into the turbine design in an attempt to make this the lightest and most efficient plant possible. The reduction gear was, as usual, three-pinion, single-reduction, double-helical, but in this case the center or top pinion was driven only by the high-pressure element of the backing turbine. Each of the side pinions was driven by two turbines mounted in tandem, all forward of the gear, with the high-pressure element and one of the two low-pressure elements on one side and the intermediate-pressure element on the other side. Each of the low-pressure casings had a four-stage reaction low-pressure astern element in the after end.

In spite of the efforts of Blohm and Voss on the Z9-16, Deschimag was chosen to build all the rest of the destroyers which were built. The next class, the Z-17-22, was similar to Z1-4 except that the cruising and high-pressure turbines were combined in one casing. This turbine was an interesting piece of design work, in fact one of the few which was as effective as it was novel. It was a double-ended machine with steam entering at the center through a horseshoe-shaped nozzle block having the cruising nozzles in one side and the nozzles for high power in the other. For low powers the cruising element exhausted to the first-stage shell of the high-pressure element, whereas for high powers it exhausted to the cross-over between the high-pressure and intermediate-pressure units. Therefore, the cruising element was working under all conditions and no difficulty due to windage heating was experienced. This design did not quite match the Z1-4 in efficiency at cruising powers, but it certainly was a large improvement on Z5-8 and apparently there were no operational or maintenance disadvantages after the initial difficulties were ironed out.

The last destroyer design (that is, the one in use at the end of the war) was the Deschimag Z23-36. It had two essential changes over Z17-22; namely, the low-pressure was made a straight center inlet, double-flow, reaction machine with a low-pressure astern element in each end, and a separate high-pressure astern casing was added in tandem with and aft of the combined cruising and high-pressure ahead turbine. The change from impulse to reaction in the low-pressure turbine is significant, but the reasons therefor are not yet discernible. Deschimag preferred to build the impulse unit, and they claimed equal or better performance with less weight for impulse. However, the Navy dictated the change and Deschimag had to comply.

The last design undertaken before the production of surface vessels was stopped in favor of intensive concentration on the submarine program was that for the "Spahkreuzer." Although none of these ships was completed, some of the machinery was finished. The arrangement differed from the Z23-36 only in that, first, the low-pressure turbine reverted to the impulse design and, second, by putting the top pinion slightly off center they were able to get enough separation with only one idler in the gear. This resulted in the smallest specific weight and space yet realized, although an increase in turbine water rate from 7.22 pounds per shaft horsepower per hour to 7.68 at normal power had to be accepted.

Concerning the surface ship machinery, the following miscellaneous details may be of interest:

For Gears.

Center drive, four bearing pinions were the vogue (two of the bearings were between the two helices).

Hardness of meshing elements was about the same as used in this country (150–175 BHN for large gears, 210–255 for pinions).

Tooth loadings were comparable to ours.

Pinions were made of silicon steel instead of nickel steel, because of the shortage of nickel at first, and later because they liked it better.

Thrust bearings were of the Michel type (similar to our Kingsbury).

All gears were hobbed, generally on Reinecker or Pfauter machines. No shaping or Sykes cutting was used. Shaving was not known.

Most ships, particularly cruisers and larger, had one or more shafts equipped with a pneumatic or hydraulic disconnect clutch to improve cruising performance and assist in isolating damage.

Hollow boring of rotating parts to save weight was universally used. although this was a short step on a long road.

For Turbines.

Buckets and blades were of familiar orthodox design. Side entry dovetails were used for short impulse blades, and circumferential dovetails of the "inside" type were used for the longer impulse and all reaction blades. Integral covers were used frequently for the shorter impulse blades, while the longer ones had separate covers attached by peened tenons. Reaction blades had no covers but were tie-wired together in groups. Impulse blades were usually long-tailed. Bucket or blade material was usually 15 per cent chrome steel.

High-pressure casings were cast, using carbon moly steel.

Low-pressure casings were usually fabricated, using mild rolled plate.

Some designs had the entire turbine assembly mounted on the condenser, which was, in turn, spring supported.

Most of the thermodynamic, aerodynamic and hydrodynamic data used by the turbine designers came from the colleges and universities. None of the manufacturers had any considerable research staff or facilities.

In contrast to the surface ship picture, the turbine and gear installation in the Walter submarine plant presented several advanced features. The Type XVII was designed to have two 2500-horsepower turbines driving a two-pinion, single-reduction gear which in turn drove a three-planet epicyclic gear of unique design. Only one turbine was installed in the boats because of too low capacity of the condensing system. The turbines were 14-stage, single-flow reaction machines turning at 14,000 revolutions per minute. The first reduction was of orthodox, double-helical design, reducing the speed to 3770 revolutions per minute from which the planetary reduced it further to 580 revolutions per minute for the propeller. It is interesting to note that this revolutionary gear arrangement resulted in a specific gear weight of 0.83 pound per shaft horsepower, whereas our most comparable unit as far as power and ratio are concerned weighs something over 5.5 pounds per shaft horsepower.

The combustion materials used for the cycle were such that the vapor supplied to the turbine inlet was about 15 per cent carbon dioxide by weight, the remainder being water vapor. The condition of the mixture was 412 pounds per square inch gage at 1022 degrees F. total temperature and the exhaust pressure was 7 pounds gage when operating on the surface. When operating submerged, the exhaust pressure was about 7 pounds gage plus the static pressure corresponding to the submerged depth of the overboard exhaust opening. There was also a small amount of free oxygen in the mixture, so the problem of corrosion resistance of the steam path was of major importance.

The most extraordinary feature of these turbines was the complete absence of a horizontal casing joint. This feature was specified in order to avoid as much as possible the danger from leakage into the submarine of the CO_2, CO and O_2 of the working fluid, although the designers insisted that such precautions were not necessary. The method of accomplishment was quite ingenious. The external casing was a cylinder with end plates which could be removed after dismantling the horizontally split bearing housings. The inner barrel

or blade rings then could be removed along with the rotor from the outer casing by removing the retaining dowels.

The blade rings were split horizontally and held together by hoops or snap rings and nothing in the design was allowed to digress from cylindrical shape in order to avoid thermal distortion. The relatively high efficiency of the unit resulted from small blade tip clearances, and these would not have been practicable in any design involving anything other than radially equal thermal expansion and contraction.

The workmanship throughout this turbine and gear installation exhibited a high degree of precision and finish. The blades were tapered and of tear-drop cross-sectional shape. The entire rotor was of stainless steel. The planetary gears had case-hardened and ground rotating parts. In general, whereas the surface ship propulsion machinery can be considered inferior to our own, this plant is believed to equal or exceed anything produced in this country, so far, for powering a ship.

Diesel Engines.

The well-developed, reliable MAN engine, which was installed in the majority of submarines, was the four-cycle, single-acting Model MV 40/46, supercharged by a Büchi turbo-blower to yield 145 pounds per square inch brake mean effective pressure at maximum rating, and weighing 22 pounds per brake horsepower at that rating. It is of conventional design, 6 cylinders, 15.7-inch bore, 18.1-inch stroke, 470 revolutions per minute continuous and 520 revolutions per minute maximum speeds, with a fuel consumption of 0.362 pound per brake horsepower at continuous rating. Pielstick-type dampers were installed originally, but eliminated for Type XXI submarine installations. Büchi turbo-blowers also were eliminated for Schnorchel installations because of increased back pressure inherent in the system.

In the high power output or larger machines, the Model Z 32/44 double-acting two-cycle type is MAN's best achievement. This is a 24-cylinder Vee engine, 12.6-inch bore, 17.3-inch stroke, delivering 415 continuous horsepower per cylinder at 565 revolutions per minute, or 500 horsepower per cylinder at 600 revolutions per minute at maximum rating. The net result is 9960 continuous horsepower at 15.6 pounds of weight per horsepower, with a maximum output of 12,000 horsepower. The continuous load rating is high for engines of this size and power; namely, 83 per cent of the maximum rating. It was planned to use six of these engines in a Diesel-powered destroyer of the *Narvik* class, the hull of which had been launched and was scuttled at the Deschimag plant at Bremen. These were arranged on three shafts, the centerline shaft having four engines connected through fluid couplings, and each wing shaft having but one engine. This engine was originally built as an in-line engine for the cruiser *Leipzig*. The general design is similar to the Hooven-Owens-Rentschler built in-line engines used on our submarines, patrol craft and other naval engines.

The same design engine, a 24-cylinder Vee-type, was also built in a 16.5-inch by 22.75-inch bore and stroke, with a maximum output of 15,192 horsepower. Based on a continuous rating of 13,800 horsepower, this engine weighs 26.9 pounds per horsepower. It had been built as a six-in-line for several years and was known as the "pocket battleship engine." Following 1800 hours of operation on a V-6 experimental unit built in 1940, the first V-24 was built and had had about 300 hours of operation on the test bed. A similar type 8-in-line engine, 25.6-inch by 37.3-inch bore and stroke, was also under construction. It had a maximum output of 1629 horsepower per cylinder but was heavier, weighing between 40 and 50 pounds per horsepower. Centrifugal scavenging pumps were used on all double-acting models.

The Daimler-Benz Model MB511 represents the outstanding four-cycle, supercharged, lightweight Diesel designed for naval craft in Germany. It is of the pre-combustion chamber type, which is pre-eminently suited to high speeds at high brake mean effective pressure. Continuous outputs at 124 pounds brake mean effective pressure with 2410 feet per minute piston speeds and maximum outputs of 150 brake mean effective pressures with 2660 feet

per minute piston speeds, are the usual supercharged performances. A test stand performance of 165 pounds brake mean effective pressure at 2840 feet per minute piston speed was obtained on a one-hour rating. The design of the engine is the product of 10 years of continuous effort to build life and high performance into a reliable marine power plant of minimum weight. As a consequence, the materials used are aluminum and steel alloys of carefully stress-analyzed and fatigue-tested forms. On the basis of continuous load rating of 1980 horsepower at 1480 revolutions per minute, the bare engine weight is 4.4 pounds per brake horsepower. Maximum load rating is 2500 horsepower at 1630 revolutions per minute. Fuel consumption of 0.396 pound per brake horsepower per hour at continuous rating was obtained. The blower, gear-driven from the drive end of the crankshaft, is provided with a hydraulically operated multi-disk clutch, which drives the compressor only when the higher loads are required. Under ordinary cruising conditions, the blower is not in use and wear and tear both on the high-speed blower (10 times crankshaft revolutions per minute) and the engine itself are reduced. No vibration damper was required on the crankshaft for Schnell-Boot installations, which were designed to avoid torsional criticals in the shafting. The supercharger blower, however, was isolated from the crank system so that any excitation from either the crank or the blower was not transmitted to multiplying amplitudes or high stress values. A spring drive is built into the crankshaft gear hub, which drives the blower train of gears.

Several development engines were also studied, which included a two-cycle sleeve-valve engine, a two-cycle port-scavenging engine, and an air-cooled rotary disk valve engine in its early stages of development. These three engines were attempts to develop high-output, high-speed engines, fitted with fuel injection compatible with whatever compression ratios might be selected to permit the burning of Diesel fuel on the one hand or benzine on the other, choice of fuel being dependent on economy, supply or war requirements. Little practical progress was made.

The principal Deutz development of interest was a 16-cylinder, horizontally opposed-piston engine being developed for compression ignition and also for benzine spark ignition for use in aircraft and in tanks. Its arrangement could prove suitable for high-powered, high-speed, amphibious vessels or PT boats. A "square" engine—6.3-inch bore and stroke—it develops 1800 horsepower at 2500 revolutions per minute at 5 atmospheres brake mean effective pressure. Maximum take-off brake mean effective pressure at 2700 revolutions per minute is 7 atmospheres. Fuel consumption is 0.35 pound per brake horsepower per hour at 1200 brake horsepower, 1500–1800 revolutions per minute. The design of the engine is excellent and shows refined features of careful planning. The materials employed are aluminum, magnesium and well-stressed alloy steels. The weight of the engine is 3000 pounds or 1.67 pounds per brake horsepower.

The Junkers Jumo 205 opposed-piston aircraft Diesel engine, which was publicized widely prior to the war, was used only experimentally. It failed to live up to its earlier expectations and consequently was not built or used in production quantities.

The development of "closed-cycle" Diesel operation (i.e., the idea of supplying oxygen to augment recycled exhaust gas to support combustion of fuel in a Diesel engine, providing a closed circuit for submarine propulsion while completely submerged) had reached the stage of satisfactory test stand performance. Operation of a conventional Daimler-Benz four-cycle engine, similar to that used in S-boats, could be changed momentarily to normal Diesel operation or vice versa without special preparation. The maximum output on "closed cycle" was comparable to that obtained on normal Diesel operation. At two-thirds of maximum load, the fuel consumption of the "closed cycle" power plant was but 23 per cent greater than for similar power on surface operation and the oxygen consumption was four times the fuel consumed by weight. A contemplated submarine installation in a modified Type XXVI submarine had a calculated cruising radius of 11½ hours at 15.2 knots submerged. The development appeared to be of secondary impor-

tance after the development of the Walter system with higher power outputs but was good insurance in case of insufficient supply of hydrogen peroxide or destruction of supply sources.

Several development trends are worthy of comparison with United States practice.

Considerable effort was expended in efforts to increase engine output per cubic inch of piston displacement. In small-bore, high-speed units, the future thoughts have been, in the majority of cases, expressed in two-cycle models. There was recognition of the value of uniflow scavenging for high-speed removal of spent gases and the rapid filling with air at higher than atmospheric pressures, with attendant opportunities for creating turbulent air flow to assist combustion. There was a desire to stay away from poppet valves in the cylinder head, and this reluctance to employ valves led experimenters to revert to the simpler structures incident to loop-type port scavenging. The hope for higher brake mean effective pressure in this latter arrangement, whether for very high or lower speeds, was voiced in the possibilities of using higher charging pressures and more efficient exhaust-gas turbo-blower applications. Supercharging was the principal avenue by which improved engine performance has been achieved.

In most modern equipment observed, the centrifugal-type supercharger was employed, either direct gear-driven or driven by means of an exhaust-gas turbine. The main objections to the Roots-type blower were weight and space factors and, to some extent, noise. Considerable thought was given to methods of reducing the noise level of centrifugal types. In all cases, isolation of torsional vibrations from either the crankshaft amplitudes or from blower excitation was obtained by the use of some spring-type of damper placed somewhere between the crankshaft and the impeller shaft.

There appeared to be a tendency to avoid piston cooling wherever possible. This was done in order not to place too heavy a burden on the piston and ring design for oil control and to maintain minimum oil consumption.

The most significant improvement in four-cycle cylinder heads of both Daimler-Benz and MAN designs was the manner of directing the water flow across the surface above the combustion chamber. A diaphragm or dividing plate was designed to cause the flow of all cooling water from the outer periphery of the jacket directly across the center of the head, and around the exhaust and inlet valves in transit.

It seemed to be almost universal practice to use Bosch fuel-injection equipment of conventional design.

All crankcase design showed very emphatically the intention of the designers to provide deep crankcase sections of high section moduli forms providing maximum stiffness to bearing alinement compatible with engine weight.

Various types of bearing metals known to American Diesel manufacturers, such as SATCO, copper-lead, babbit and aluminum, were also used in Germany. Prevailing practice appeared to be the use of narrow bearings on stiff, short crankshafts. MAN believed aluminum tin bearings were best for high unit loading, but it was highly essential to get oil to the bearing immediately upon starting.

Great faith was placed in the value of laboratory fatigue and dynamic and static stress studies on crankshafts. MAN, together with the Navy, invested four million marks in a laboratory largely composed of crankshaft testing equipment. MAN engineers had preserved a considerable mass of evidence in field experience to justify laboratory testing techniques and establishing stress concentration and damping factors for new crankshaft determinations.

Considerable fuel testing—both Diesel and gasoline—was carried on. The German Navy used good gas oils for their submarines, but these were not too plentiful. It was necessary, therefore, for manufacturers to use various blends of fuels, some of which contained tar oils. Some synthetic fuels were available, and various blends were supplied without knowledge of the purchaser. Much work had been done on cetane improvers, but primarily for the purpose of assisting starting of low cetane fuels. The Fischer-Tropsch

high cetane fuels were considered too high in hydrogen, requiring more oxygen to burn them.

No detergent types of lubricating oils were used in Germany. Those used for Diesel engines were generally a close-cut high viscosity index oil in the petroleum range, and some excellent high stability synthetic products for aircraft uses. MAN found the synthetic oils to be more stable between oil changes and suffered less deterioration in the Diesel engine under high temperatures than straight mineral oils.

APPENDIX 1.

A considerable amount of detailed information on matters discussed here which could not be included in a paper of this scope is available in the technical reports of the Naval Technical Mission in Europe. These documents, which were transmitted to the Office of Naval Intelligence, have been reproduced and groups of them have been declassified from time to time.

In the list which follows will be found the titles and identifying numbers of some of the more valuable of these reports which formed the basis of this paper and from which further information may be obtained.

The availability of these reports to individuals varies with the classification. Those which are unclassified have all been transmitted to the Publications Board (Mr. John Green), Department of Commerce, Washington, D. C., and can be ordered through this office. Lists of titles and reproduction costs appear in the weekly "Bibliography of Scientific and Industrial Reports" compiled by the Superintendent of Documents, Government Printing Office. The remainder of the reports are still classified *restricted*, except in one case, and these reports remain in the possession of the Office of Naval Intelligence and are closed to the general public. Persons who have Government business which involves studying this type of material may obtain clearance by justifying the necessity to their personal contacts in the Navy Department. It is also possible to be cleared by the Foreign Technical Intelligence Desk (Op-23F2) of the Office of Naval Intelligence.

Unclassified Reports

NavTech MisEu No.	Title
70-45	Diesel Engines
114-45	Diesel Engine Research and Development
137-45	Boilers
138-45	Turbines
147-45	Internal Combustion Engines
358-45	Underground Factories and Storage Depots
377-45	Experimental Diesel Engine
379-45	Survey of Diesel Engine Developments
382-45	Water Tunnels at Gottingen and Braunschweig
442-45	Main Propulsion Turbines and Gears
509-45	Steam Turbine Power Plants
535-45	Research and Facilities of Foreign Model Basins
543-45	Facilities, etc., Hamburg Tank

Restricted Reports.

69-45	E boats
94-45	Ingolene Cycle for Submarines
159-45	*Prinz Eugen*
214-45	Closed Cycle Diesel Development
216-45	Naval Gas Turbine Developments
222-45	Loss of Battleship *Tirpitz*
224-45	*Bismark* and *Tirpitz*
251-45	Shock and Sound Insulation for Submarines
275-45	Ceramic and Water-cooled Turbine Blading
303-45	Submarine Main Propulsion Equipment
305-45	Hydraulic Systems on Submarines

309-45 Ventilation, Air Purification and Oxygen on Submarines
310-45 Batteries and Battery Ventilation on Submarines
312-45 Submarine Design, 1935–1945
322-45 Kreislauf Cycle
338-45 Schnell Boot
357-45 Main Propulsion Gearing
403-45 Influence of High Submerged Speed on Submarines
477-45 Submarine Construction Organization
526-45 Projected Battleships, Types O and H
542-45 Submarine Propulsion Plant Using Ingolene Boiler

CONFIDENTIAL REPORT.

162-45 Walter Submarine Machinery

HYDROGEN PEROXIDE FOR PROPULSIVE POWER—PRODUCTION AND USE BY THE GERMANS DURING WORLD WAR II.

Brief mention is made in the previous article of the Walter Turbine which was developed by the Germans for submarine propulsion during World War II. This article contains a more complete discussion of the utilization of hydrogen peroxide for power production and points out some of the difficulties involved. The article was prepared by Captain Logan McKee, U.S.N., Director of Ship Design, Bureau of Ships, and is reprinted from the December 1946 issue of "Mechanical Engineering."

The use of hydrogen peroxide for propulsion purposes has been referred to from time to time in the public press. Certain details of the applications have of necessity been restricted, and similar restrictions still apply to some phases of the subject. However, there are features of the use of this material for power production, particularly by the Germans during the last war for V-bombs and for propelling airplanes and submarines, which are not restricted, and which it is felt would be of engineering interest.

CHEMISTRY INVOLVED.

The formula for hydrogen peroxide is H_2O_2, and the material we will deal with is further defined by giving the per cent concentration, i.e., it is normally manufactured at about 27 per cent concentration. If a higher concentration is required, it is distilled under low absolute pressure, starting at about 50 mm Hg and decreasing the pressure to about 30 mm as the concentration is increased to 85 per cent. When used as a disinfectant the concentration is about 3 per cent, the remaining 97 per cent being water.

The Germans used a concentration of 80 to 85 per cent for 26 different war weapons and had it in an experimental stage for 40-odd others. They gave it a cover name of "Ingolin." Professor Walter, who was the foremost proponent of its use in Germany, named it after his eldest son, Ingol. It is also known there as "T-stoff."

It was sometimes used as a monofuel or primary fuel, i.e., using only the heat of dissociation of the H_2O_2 into H_2O plus O_2. The heat of dissociation for 80 per cent concentration is roughly 1000 Btu. per lb. The mixture, in that case, would be 80 per cent H_2O vapor (superheated steam) and 20 per cent O_2 by volume (63 per cent H_2O and 37 per cent O_2 by weight), and the mixture would be at 500 C. (932 F.), assuming that the liquid was supplied at 70 F. It was, however, used more often as a secondary fuel, i.e., all or nearly all of the free oxygen was further burned with a fuel, such as decalene (which is similar to Diesel oil) or methyl alcohol. The reaction would, in some applications, take place all in one step by the use of a liquid catalyst and fuel combined. The Germans named the liquid catalyst and fuel mixture they used "Helmann" after Walter's second son. Their mixture was hydrazine hydrate and methyl alcohol. The reasons for using the material as a secondary

fuel is to take advantage of the larger heat release per pound of mixture and also because the dollars and cents cost per Btu. obtained from fuel is much less for that than from H_2O_2.

It will be noted from the foregoing that the heat of dissociation of H_2O_2 per pound is about 1/19 of that of the heat of combustion of a pound of oil. The value of H_2O_2 is therefore its ability to furnish free O_2 when and where you want it. It has advantages over liquid or gaseous oxygen in that it is more easily contained and it produces a lower flame temperature.

Production of H_2O_2 in the United States.

Prior to and during this war, H_2O_2 in this country was not concentrated for commercial or military purposes at greater than 50 per cent, although some laboratory tests had taken it higher. The reason it had not been taken to higher concentrations is that, unless certain rules are known and carefully observed, it is violently unstable and, naturally, the danger increases with the concentration. It follows that the Germans were far more advanced in the subject than any other country. We have been interested in it in this country for years, but did not reach the point of taking advantage of its characteristics in the higher concentrations. We now can and do concentrate it commercially to a higher percentage than the Germans ever did and to a greater degree of purity. For example, the Buffalo Electro-Chemical Company will supply it at 90 per cent concentration with only 5 parts per million impurities. Du Pont can do about the same. The Germans usually had much greater amounts of impurities, some of them not very harmful.

The foregoing will give an idea of the order of the concentrations used.

Characteristics of Hydrogen Peroxide.

In the concentration used by the Germans, the specific gravity is 1.37 at 32 F. It is colorless or slightly yellow and has a distinctive, but not unpleasant, odor. The 100 per cent concentration freezes at 20 F., and boils at 306 F. at atmospheric pressure, and its specific gravity at 32 F. is 1.46 plus. A peculiar thing is that 80 per cent H_2O_2 freezes at 11 F. and, of course, 1 per cent at just under 32 F.; 60 per cent solution freezes at —70 F. The boiling point at atmospheric pressure has to be obtained by extrapolation as the material detonates before reaching the boiling point. Concentration is determined usually by taking its specific gravity, although titration gives a more accurate check. Impurity in the form of sulphuric acid throws off the determination of concentration by observations of the specific gravity and this gave the Germans so much trouble that they learned to eliminate the H_2SO_4.

After a stabilizer (or stabilizers) is added, (the Germans used oxyquinoline or phosphoric acid) it can be left in an open container without too rapid a loss in concentration. Concentration drops off rapidly at first and becomes progressively slower as time goes on, provided no unusual conditions are encountered, and provided further that the heat of decomposition is allowed to dissipate. The drop during the first month in storage is about equal to the drop to be expected for the remainder of the year. It is roughly 1½ per cent the first month and 3 per cent the first year and very little thereafter, but may be much less.

Provided one's hands are clean, they can be put into a high concentration of H_2O_2 without any immediate sensation except a slight prickling. In less than one hour, however, the hands will look as though whitewash had dried on them. When that coating wears off, the skin will be bleached white. All except a few materials act as catalysts for H_2O_2, so if there is dirt under the fingernails or on the hands, a burn will result. The antidote is water, and the way it acts is to reduce the concentration quickly.

With regard to materials used for handling and storage, iron and copper must be carefully avoided as they are catalysts. Stainless steel, 99.6 per cent pure aluminum, glass, ceramics, and certain synthetics can be used; the most widely used synthetic in Germany being polyvinyl chloride, which looks like rubber in its usual form, although it is also available as a cloth. It is used as a packing material for valve stems and for gaskets but it has one serious

deficiency in that it shrinks and hardens with age. It is therefore difficult to keep the systems tight when using it. It is understood that du Pont has a much superior synthetic for the purpose.

Precautions in Handling H_2O_2.

When the Germans first started handling and storing concentrated H_2O_2, they used many precautions which they later determined from experience to be unnecessary. The requirements which are essential are carefully observed. They have no more fear of handling it than they have for handling gasoline, for example.

In storage the first indication of instability of H_2O_2 is a rise in temperature. If something isn't done about it, the condition becomes progressively worse, finally reaching the danger point when the temperature registers 140 F. At that time, the H_2O_2 is dumped into water. In the early storages, there were cooling coils in the tanks and a sprinkling system above them, but experience showed that they were unnecessary. On noting a rise in temperature, more stabilizer is added.

By far the most fundamental rule in handling concentrated H_2O_2 is to have fresh water instantly available for diluting any spilled H_2O_2. Almost without exception, when concentrated H_2O_2 is spilled, a fire starts. All that is needed to put it out is to reduce the concentration below 78 per cent, provided, of course, that no large amounts of combustibles are in the area. The reason that the fire starts so readily is that there are usually some small amounts of combustibles present, such as grease, paint, wood, or cloth on which the H_2O_2 spills; the dirt on the combustible materials acts as a catalyst; heat of dissociation of the H_2O_2 raises the temperature of the combustibles to their ignition point, and the free O_2 is available to maintain the combustion. A little dilution of the H_2O_2 keeps the heat of dissociation below the autoignition temperature of the combustibles. The Germans kept water in the bilges of the engine rooms of their H_2O_2-driven submarines to reduce the concentration of any spilled H_2O_2.

The men who handle concentrated H_2O_2 wear scrupulously clean polyvinylchloride coveralls (which don't look much different from any other kind of coveralls), boots and gloves of the same material, except that the latter are in the form of synthetic rubber.

Laboratory demonstrations of the properties of concentrated H_2O_2 are quite impressive, two in particular. In the first, a clean piece of wood has one end submerged in an open container of H_2O_2. No reaction takes place when it is withdrawn. Another piece of similar wood is rolled on the floor and then dipped in the H_2O_2. It catches fire upon being withdrawn. The dirt it picks up from the floor contains materials which are catalytic. In the other demonstration, a small amount of liquid catalyst (hydrazine hydrate) is put in a large flat pan. Then the chemist stands off as far as possible and throws about 2 oz. of H_2O_2 from a beaker into the pan. Only a small part of it actually gets in the pan, but what actually arrives reacts with a loud noise and eruption of sparks. In other words, the decomposition of H_2O_2 in the presence of a catalyst is very violent. In fact, it is a more powerful explosive than TNT, if properly combined with catalyst and fuel; that is, it contains more releasable energy per pound. The material mixes in all proportions with alcohol and with glycerine. It is many times more dangerous to have around than dynamite or TNT, as it is much more unstable.

Hydrogen-Peroxide Manufacture.

The usual method of manufacturing H_2O_2 is the hydrolysis of persulphuric acid, but it can be manufactured by any one of several chemical methods. Quite a lot of electrical power and expensive facilities are required for the hydrolysis method so the product is expensive. The cost will come down considerably but it will probably remain high as compared to fuels or gaseous or liquid O_2. Toward the end of the war, Germany had a capacity for manufacturing about 3000 tons a month which was not nearly the amount she wanted. It required the services of about 3000 men to do the manufacturing.

Their largest plant was well hidden in the Harz Mountains and was not located by our Intelligence until after the war.

The utmost care is used in the manufacture and cleaning of storage tanks. They are gas-welded and great pains are taken to leave no possibility of H_2O_2 seeping into crevices or under the edges where plates are lapped. Welding rods are carefully selected in order to obtain alloys free of copper. An oxidizing flux is used. After welding the interior of the tank is cleaned and polished, treated with 10 per cent nitric acid, washed, painted (or filled) with 20 per cent solution of caustic soda, washed, treated with 10 per cent nitric-acid solution, washed, then the inside coated with wax.

The wax is composed of parafin, beeswax, and a synthetic prepared by I. G. Farben. The principal purpose of the wax is to fill any small holes or crevices which may exist. The outside of the tank is heated to melt the wax being applied to the inside. The reason for taking such precautions to fill crevices is that several explosions were attributed to H_2O_2 becoming trapped between plates where they overlapped.

The shape of the tank is influenced by trying to get as small a surface-volume ratio as possible. After the tanks are ready, H_2O_2 is put in and carefully watched for 2 weeks. If undue decomposition is noted, the tank is emptied and the interior again treated. About 10 per cent of the interior is left empty, as a gas space. All piping and valves used for transferring H_2O_2 must be completely flushed with fresh water after each use.

German Uses for Hydrogen Peroxide.

The Germans used H_2O_2 to launch V-1 bombs, to drive fuel pumps in the V-2 bombs, to drive torpedoes, ME163 and 262 airplanes, and submarines.

For the V-1 and V-2 bombs only the heat of dissociation was used. It was a rather expensive use of energy but was so positive in action that its simplicity and reliability were worth the cost.

Airplanes. The performance of the airplane was astounding. The thrust from the rocket engine corresponded to that of a 3700-Hp. conventional-type engine, and it weighed about 250 lb. It was not subject to the explosion danger to nearly the extent of the torpedo engine, although the same principle of H_2O_2 in combination with a catalyst and fuel was used. The working fluid was not put through a turbine in this case, however, so power was obtained entirely by jet action. It could be cut on and off instantly.

It was related to the author by a German that a woman made the first test flight in the plane. The fuel supply lasted only a very few minutes but thrust was so great that the plane could not be flown in level flight, with the engine on, for more than a few seconds. It very quickly approached the speed of sound. It could climb to 30,000 feet in 2 minutes, which made it extremely valuable as an intercepter plane. The first planes, the ones which saw war service, had only a main jet so the engine was either full-out or dead. A later design incorporated a cruising jet, in addition to the main, which increased the time that the engine could be kept on about twentyfold. Its rate of climb and speed in level flight, under cruising jet conditions, were not much better than that of an orthodox gasoline-engined fighter plane, and its radius of action much less; but, until the fuel was exhausted, it had a tremendous reserve of power.

Torpedo Drive. The torpedo drive was somewhat similar to the submarine drive. In the former, a liquid catalyst and fuel combined (Helmann) were used. After 2 or 3 seconds, decalene was admitted and the Helmann shut off. Decomposition of H_2O_2 was then accomplished by heat. Temperature in the combustion chamber must remain above 1800 F. The catalyst chamber and combustion chamber were combined. That resulted in a rather dangerous situation, as the arrival of the three liquids had to be accurately co-ordinated. If any was off-time a serious explosion resulted, and our information indicates that about one out of each 100 torpedoes was wrecked that way. The performance characteristics were much superior to those of any other torpedo and they were essentially wakeless.

Submarine Propulsive Power.

In 1943 the Germans realized that our antisubmarine forces had won against the type 7C submarine which they had depended on up to that time. The 7C was fundamentally much like the ones we used in characteristics, except that it was considerably smaller so its speed and endurance were less. It displaced about 700 tons on the surface, had a surface speed of 17 knots and a submerged speed of about 7.5 knots. The Germans were able to change their thinking radically in regard to submarines. They realized that, in order to cope with our antisubmarine forces, they would have to go to a true submarine, instead of a surface ship which could be submerged, and that, to make an attack and then escape, it would be necessary to have high underwater speed. Their first and most important development was the underwater breathing tube or "Schnorchel." They used this against us effectively. It permitted them to make a war patrol without once coming to the surface. It was essentially a pipe, which extended 18 inches or so above the surface, through which they obtained air for the Diesel engines and for ventilation.

Then, in the summer of 1943, they conscripted a group of eminent designers and technicians and assembled them in a little town in the Harz Mountains. They produced the much discussed type 21 boat. It was a completely different submarine. It would have been very effective against us had the Germans not made one fatal error in their design, as they have done so often. Their hydraulic system was so complicated that they couldn't get it to work before the war ended, although they had built about 120 of these boats by that time. Type 21 had a surface displacement of 1600 tons, a surface speed of 15.5 knots and a submerged speed of 16.5 knots for one hour. It obtained its high underwater speed by the use of large battery power and a battery with very thin grids, 0.160 inches thick.

At this time Walter, at Kiel, was able to speed up the acceptance of his hydrogen-peroxide submarine. He had experimented on the use of hydrogen peroxide since 1935 and had built and operated an 80-ton experimental H_2O_2 submarine before the war. It performed in accordance with his design and realized some 25 to 26 knots submerged speed. During the war, he supervised the design of four school boats, the first of which went into operation in the fall of 1943. They were known as type 17 and performed in accordance with expectations. They were followed by five operational boats, type 17B. These were 380-ton-surface-displacement vessels, surface speed 8.5 knots, submerged speed 25 knots, and had two bow torpedo tubes and two spare torpedoes. They never had war patrol because there was never enough hydrogen peroxide to permit them to so operate.

The last word in submarine design was to have been the type 26 Walter boat. It was under construction when the war ended. It was to have been of 900 tons surface displacement, 11 knots surface speed, and 24 knots submerged speed. It had 10 torpedo tubes but no spare torpedoes. The hydrogen-peroxide engine (it had only one shaft) was rated at 7500 Hp. Our bombing force never permitted the Germans to get the engine assembled.

However, the author collected together all the parts for one engine in the summer of 1945 and shipped them to England, where the engine now is. A schematic diagram of this engine is shown in Figure 1.

Details of Submarine H_2O_2 Power Plant.

There is a so-called triple-feed pump which pumps H_2O_2, Diesel oil, and water. The Ingolin pump parts, piping, and catalyst chamber are stainless steel. The first position on the starting wheel allows only Ingolin to be pumped. Ingolin reaches the catalyst chamber where it sprays on porous porcelain stones on which are fixed calcium, potassium, or sodium permanganate. As mentioned before, the Ingolin breaks up into steam and O_2; 80 per cent steam by volume, at a temperature of about 930 F. From that point on, materials are simple alloy steels. The steam and O_2 mixture go to the combustion chamber. Soon after it reaches that point, the cooling water is allowed to circulate. The water makes two passes through the combustion-chamber water jacket and then sprays into the turbine working fluid. Soon

FIGURE 1.— DIAGRAM OF COMPLETE-CYCLE ENGINE USING HYDROGEN PEROXIDE.

after the cooling water is admitted, decalene is allowed to spray into the combustion chamber.

The temperature in the combustion chamber is above the autoignition point, but an automobile spark plug in the combustion chamber is energized to assure ignition. Danger of explosion exists in the event ignition fails. A proportioning device admits water, Ingolin, and decalene in a ratio of approximately 12 to 9 to 1. The proportioning device gave more trouble than any other item in the machinery plant. The earlier ones were installed on the triple-feed-pump suctions and the later ones on the discharges. There is more hazard connected with those on the discharge side because when the flow of Ingolin is restrained a pressure is imposed upon it and it is churned in the Ingolin pump.

Flame temperature in the combustion chamber is about 4000 F. The water is sprayed into the combustion chamber in an amount to reduce the temperature to the desired degree. The Germans used approximately 1020 F. for their turbine working fluid. Working fluid is now about 94 per cent steam by volume and 85 per cent by weight. It goes through what amounts to a conventional steam turbine, operating at 14,000 Rpm., and into a contact-type condenser where the steam is condensed.

In the earlier engines, the noncondensing gases were forced overboard by the turbine back pressure but in the engine for the type 26 boat, they were pumped overboard by a Lysholm-type positive-displacement rotary compressor. The spray water for the contact condenser is cooled by circulation through a surface-type heat-transfer unit located outside the pressure hull; sea water is forced through the heat-transfer unit by scoop action.

The machinery installation is not quite as simple as the sketch indicates. The turbine has the same problem of shaft-gland sealing as any other steam turbine. It is water-sealed with a gland leak-off and leak-off condenser. There are three auxiliary lubricating-oil pumps which are used for starting only. When running, lubrication is accomplished by pumps driven from the gears. There is a Diesel engine for surface and Schnorchel operation; a main motor and a creeping motor for submerged battery drive. The main motor also serves as a generator while the boat is being driven by Diesel engine or Walter engine. It furnishes power for auxiliaries and for charging batteries. The creeping motor is a high-speed low-power affair which drives the main shaft through multiple V-belts. The main turbine drives through a conventional single-reduction double-helical gear and then through a planetary gear. The Diesel and main motor are on the same shaft and drive only through the planetary gear. The single-reduction gear and turbine can be disconnected

by a clutch. The main motor and Diesel or just the Diesel are connected or disconnected by clutches. The main motor, however, always operates, either as a motor or a generator, except when the creeping motor is in use. All gears are disconnected when using the creeping motor.

Germans Complicated H_2O_2 Machinery Unnecessarily.

The machinery was subject to many breakdowns, the principal reason being that in Germany a design error, or what might be interpreted by the courts as a design error, was a civil offense. That caused the designers to guard against all damaging casualties to machinery, instead of accepting some hazards for the sake of simplicity. For example, there were five shut-down devices installed to guard the turbine. That resulted in complications which led to frequent malfunctioning of safety devices. The machinery compartment was sealed off when machinery was running but the crew would go in with little hesitation when casualties occurred.

Submarine propulsion was the ideal application for H_2O_2, its value resulting from the fact that using H_2O_2 is a good way to obtain O_2 for combustion when operating submerged. By "a good way" is meant that it had many advantages. It was carried in collapsible polyvinyl-chloride bags outside the pressure hull. The hydrostatic pressure outside kept a positive pressure on the Ingolin pump. Ingolin used from inside the bags was displaced by sea water outside the bags, which compensated for about ¾ of the weight of Ingolin used, the difference being the difference in the specific gravity of the two liquids. Fresh water obtained by condensing the H_2O in the turbine working fluid could compensate for the difference, in order to maintain neutral buoyancy. Actually it was not done that way but was accomplished by admitting sea water into a compensating tank. Excess fresh water was forced overboard with the CO_2 and other noncondensing gases.

The one big disadvantage to the use of Ingolin is its cost. As indicated before, about 9 times more of it is used than Diesel fuel, and it costs 60 or 80 times as much per pound. Multiply 60 or 80 by 9 and the result is high. As a matter of fact, its cost is near enough to a thousand times greater than fuel oil to use that figure as a rough comparison of costs when using H_2O_2 and oil for producing power.

A MARINE GAS TURBINE INSTALLATION.

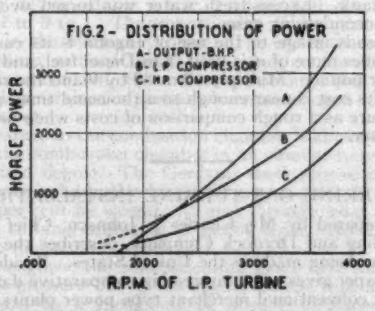
This article, prepared by Mr. Charles J. Johnson, Chief Engineer of the Federal Shipbuilding and Drydock Company, describes the first marine gas turbine installation being made in the United States. Besides describing the installation, the paper gives some interesting comparative data between a gas turbine and other conventional merchant type power plants. This reprint is a copy of an abstract as published by the "Pacific Marine Review" in June 1946. The original paper was presented to the New York Metropolitan section of the Society of Naval Architects and Marine Engineers.

Upon request of the Maritime Commission, the Federal Shipbuilding and Dry Dock Company undertook the preparation of plans and material orders for a proposed installation of an Elliott gas turbine. Subsequently work was increased to include preparation of specifications and performance of all work involved in effecting a complete installation in a Liberty Ship Collier, known as the EC2-S-AW1 type. The vessel to be used for the installation was built by the Delta Shipbuilding Company, and is equipped with a standard Liberty ship reciprocating engine installation, which will be entirely replaced with the gas turbine plant.

Efficiency and Performance.

Before proceeding with a description of the proposed installation I wish to point out that published values of thermal efficiencies of gas turbines are in most cases not directly comparable with thermal efficiencies of steam turbine

installations without adjustments to comparable conditions. Published gas turbine efficiencies are usually based on low heat values of fuel, often without a statement to that effect, whereas marine engineers in this country always use high heat values in their calculations. I have noted also that the efficiencies of gas turbine units, exclusive of independently driven propulsive auxiliaries and cooling losses, are sometimes used in comparisons with complete heat balances of steam turbine plants. Such practices are misleading and are detrimental to the development of the gas turbine.



The estimated general performance of the installation here discussed for average service conditions at sea gives a fuel rate of the propulsion plant exclusive of ship's service requirements under normal conditions as .437 pound per Shp., and a thermal efficiency based on 19,600 Btu. heat value of the fuel as 29.7 per cent. The average thermal efficiency of a number of steam turbine propulsion plants designed for operation with steam at 450 pounds pressure and 750 degrees F. temperature is 24.3 per cent. The indicated advantage in efficiency of the gas turbine over what may be termed a standard steam turbine plant is therefore 22 per cent.

We must bear in mind that the efficiencies of the various components of steam turbine plants are well established by actual service performance, whereas for the gas turbine plant we must rely on laboratory and shop tests, and certain Houdry process plant experience which can be applied. There is no marine service experience with gas turbine equipment available at this time. Moreover the over-all performance of the gas turbine is affected to a greater degree by variations in efficiencies of components than is the case of

the steam turbine. For the installation under discussion, for instance, a variation of 1 per cent in compressor efficiency will result in 1.2 per cent variation in over-all efficiency, and a variation of 1 per cent in turbine efficiency will cause a variation of 1.48 per cent in over-all efficiency.

We must bear in mind also that the efficiency of the gas turbine given in the foregoing can be realized only by employing temperatures greatly in excess of steam temperatures heretofore used in either land or marine power plants. It cannot be stated that sustained satisfactory performance of the materials subjected to the high temperatures involved is assured. It is to be hoped that manufacturers of gas turbine equipment will soon be at liberty to publish complete data relating to the physical characteristics of the alloys they employ in their designs. Satisfactory materials for use under high temperatures are well worth striving for, not only to achieve improved economy resulting from higher temperatures, but also to make possible simpler power plant installations, which is an important feature of the gas turbine.

It is expected that reasonably good efficiency will be obtained at part loads. This will be accomplished by segregating the controlling and driving turbines; the provision of one stage of reheat, by means of which constant gas temperature to the driving turbine can be maintained; and by the use of a regenerator. A comparison of part load efficiencies of the unit here discussed with that of a typical steam turbine plant is shown in Figure 1.

The division of power between the two turbines at varying Rpm., that is, the power delivered to each compressor plus the useful power delivered at the low pressure turbine coupling based on Rpm., is illustrated in Figure 2.

Fuel.

Diesel fuel oil will be used for all purposes for this installation. It is understood, however, that the development of equipment which will permit the use of Bunker C oil has advanced to the point where its use can be predicted. Considerable attention is also being given to the use of coal. It is fully realized by engineers interested in the development of marine gas turbines that the use of low-cost fuel is an important factor in their successful application.

Machinery Weight.

Type	Weights	
	Dry Tons	Wet Tons
Gas Turbine.....	332	339
Steam Turbine.....	366	399
Reciprocating Engine.....	417	445

Improved plant efficiency and increased revolutions of rotating equipment have resulted not only in decreased weight but also in decreased space requirements. It is significant that the plant here discussed not only (as shown in table herewith) weighs less than a steam turbine installation of comparable power but the propulsion unit occupies 23 per cent less cubic and 45 per cent less floor area. It would appear, therefore, that machinery compartments of ships designed for gas turbine installations should be considerably less than permitted by existing rules, and that the economics of the situation will eventually lead to rule modifications whereby the benefit of machinery space saving can be realized.

Machinery Equipment and Arrangement.

The installation herein described is considered experimental in two respects: (a) the use of a gas turbine propulsion plant in a merchant ship, and (b) the use of controllable pitch propeller. The propulsion unit is, I believe, the first installation of its type, although there are or soon will be similar installations in naval vessels. Controllable pitch wheels have been used extensively by the Navy for landing craft and on a few vessels of other types. They have also been used by European builders. In this country, however, their use on merchant ships is, as far as I know, limited to one installation on a C1-M-AV1

List and Particulars of Machinery.

Item No.	No. Req.	ITEM	MANUFACTURER OR SUPPLIER	TYPE	RATING, ETC.
		HULL ENGINEERING AND FITTINGS			
1	1	Steering Gear	Struthers-Wells	Electric-Hydraulic	70 deg. in 30 min.
		All other items of Hull Engineering and Equipment will be as originally installed in the vessel.			
		MAIN PROPULSION MACHINERY			
2	1	Main Turbine—HP	Elliott	Reaction	2540 bhp max.
3	1	Main Turbine—LP	Elliott	Reaction	5172 bhp max.
4	1	Air Compressor—LP	Elliott-Lysholm	Rotary-Displacement	Capacity max. 24,366 cffa/min., inlet pressure 0.26 psi, discharge pressure 27.64 psi
5	1	Air Compressor—HP	Elliott-Lysholm	Rotary-Displacement	Capacity max. 24,366 cffa/min., inlet pressure 27.14 psi, discharge pressure 74.98 psi
6	1	Regenerator	Harrison	Extended Surface	Gas: 112,968 lb/hr, inlet temp. 963 F, outlet temp. 504 F Air: 110,160 lb/hr, inlet temp. 261 F, outlet temp. 789 F
7	1	Inter Cooler			Air: 111,600 lb/hr, inlet temp. 310 F, outlet temp. 101 F Water: 600 gpm, inlet temp. 80 F, outlet temp. 102 F
8	1	Sealing Air Cooler	Elliott		314 cffa/min.
9	1	Combustion Ch., HP	Elliott		Oil 920 lb/hr, temp. air inlet 789 F, temp. gas outlet 1335 F
10	1	Combustion Ch., LP	Elliott		Oil 493 lb/hr, temp. gas inlet 1120 F, temp. gas outlet 1400 F
11	1	Starting Motor	Elliott		100 hp. 1750/900 rpm. 115 volt d-c
12	1	Emergency Propulsion Turbine	Elliott	Impulse	300 bhp
13	1	Intake Filter	Elliott		
14	1	Intake Silencer	Elliott		
15	1	Main Reduction Gear	Farrel-Birmingham	Double Reduction Double Helical	3300 shp max.
16	1	Turning Gear, H.P. Turbine	Elliott		1 hp
17	1	Turning Gear, L.P. Turbine	Elliott		1 hp
18	1	Turning Gear Reduction Gears	Farrel-Birmingham		5 hp
		SHAFTING, SHAFT BEARINGS AND PROPELLER			
19	2	Line Shaft	Morgan Smith	Hollow Forged Steel	O.D. = 13⅛ in., I.D. = 6¼ in.
20	1	Propeller Shaft	Morgan Smith	Hollow Forged Steel	O.D. = 17 in., I.D. = 6¼ in.
21	3	Pitch Control Shafts	Morgan Smith	Solid Forged Steel	Diam. = 6½ in.
22	3	Bearings—Line Shaft	Federal	Ring Oiled	
23	1	Bearing—Main Thrust	Farrel-Birmingham	Kingsbury 6 shoe	Max. thrust = 81,000 lb.
24	1	Stern Tube	Federal	Cast Steel	
25	1	Propeller	Morgan Smith	4 Blade, Controllable Pitch	Diam. = 17 ft. 6 in.
26	1	Servo Motor Control	Westinghouse Air Brake	Pneumatic	
27	1	Prop. Hub Equalizer	Morgan Smith	Cast Steel	Diam. = 14½ in.

NOTES.

List and Particulars of Machinery—Continued.

List and Particulars of Machinery—Continued.

Item No.	No. Req.	ITEM	MANUFACTURER OR SUPPLIER	TYPE	RATING, ETC.
		HEAT TRANSFER EQUIPMENT			
28	2	L.O. Coolers (Gas Turbine)	Davis	Single Pass Straight Tube	Cooling surface 207.6 sq. ft., oil flow 72 gpm, water flow 175 gpm, oil temp. 130-105 F, water temp. 85-90 F
29	2	L.O. Coolers (Reduction Gears)	Davis	Single Pass Straight Tube	Cooling surface 131 sq. ft., oil flow 80 gpm, water flow 150 gpm, oil temp. 125-110 F, water temp. 85-89 F
30	2	L.O. Coolers (Diesel Generators)	Ross	Single Pass Straight Tube	20 sq. ft. cooling surface
31	2	Fresh Water Coolers (Diesel Generators)	Ross	Single Pass Straight Tube	60 sq. ft. cooling surface
32	2	Lub. Oil Heaters	Davis	Straight Tube	Heating surface 4.8 sq. ft.
33	1	Aux. Condenser	Originally installed	Two Pass	Cooling surface 700 sq. ft., steam condensed 15,000 lb/hr @ atmospheric pressure
		PUMPS			
34	1	Circulating—Main	De Laval	Vert., cent., M.D.	1500 gpm, total pressure 15 psi, 20 hp motor
35	2	Circulating—Dsl. Gen.	Allis-Chalmers	Hor., cent., M.D.	120 gpm, total pressure 25 psi, 3 hp motor
36	1	Fire and Bilge	De Laval	Vert., cent., M.D.	400 gpm, discharge pressure 125 psi, 50 hp motor
37	1	Fire and General Serv.	Worthington	Vert., Simplex, Steam	400 gpm, discharge pressure 125 psi
38	1	Clean Ballast	Originally	Hor. rotary, geared turbine drive	2500 gpm, total pressure 35 psi
39	1	Bilge	De Laval	Vert., cent., M.D.	425 gpm, total pressure 25 psi, 10 hp motor
40	1	Fuel Oil Transfer	Worthington	Vert., duplex steam	200 gpm, total pressure 35 psi
41	2	Fuel Oil Booster	Elliott	Gear, M.D.	10 gpm
42	3	Fuel Oil Service (Gas Turbine)	Elliott	Gear, M.D.	5 gpm
43	1	Fuel Oil Service (Boiler)	Best-Quimby	Gear, M.D.	250 gph, discharge pressure 40 psi
44	1	Priming	Nash	Hyd-Vacuum, M.D.	21 cfm, vacuum 15 in. Hg
45	1	L.O. Service (Gas Turbine)	De Laval-IMO	Vert, screw, M.D.	100 gpm, total pressure 48 psi, 5 hp
46	1	L.O. Service (Reduction Gears)	De Laval-IMO	Vert, screw, M.D.	100 gpm, total pressure 48 psi, 5 hp
47	1	L. O. Standby (Gas Turbine)	Worthington	Vert., simplex, steam	100 gpm, total pressure 48 psi
48	1	L.O. Standby (Reduction Gears)	Worthington	Vert., simplex, steam	100 gpm, total pressure 48 psi
49	1	L.O. Booster	De Laval-IMO	Hor., screw, attached	15 gpm, discharge pressure 90 psi, spare pump furnished
50	2	Compressor Jacket Circulating	Elliott	Hor., cent., M.D.	15 gpm, total pressure 5 psi, ½ hp
51	1	Condensate	De Laval	Hor., cent., M.D.	30 gpm, discharge pressure 25 psi, 1½ hp
52	1	Boiler Feed	Worthington	Vert., simplex, steam	40 gpm, discharge pressure 200 psi
53	1	Feed Injector	Schutte-Koerting	Double tube, steam	40 gpm, discharge pressure 200 psi
54	2	Drinking and Washing Water	Originally installed	Hor., duple, M.D.	10 gpm, total pressure 60 psi ¾ hp
55	1	Sanitary	Originally installed	Vert., simplex, steam	140 gpm, total pressure 60 psi

NOTES.

List and Particulars of Machinery—Continued.

Item No.	No. Req.	ITEM	MANUFACTURER OR SUPPLIER	TYPE	RATING, ETC.
		OTHER EQUIPMENT			
56	1	S.W. Evap. Feed	Worthington	Hor., duplex, steam	20 gpm, total pressure 50 psi
57	1	Distilled Water Distributing	Worthington	Hor., duplex, steam	8 gpm., total pressure 15 psi
58	2	Hydraulic Oil		Hor., gear, M.D.	28 gpm, total pressure 300 psi, 7½ hp
59	1	Hydraulic Oil Tank	S. Morgan Smith	Cyl. steel plate	600 gal, 300 psi
60	1	Boiler	Foster-Wheeler	"D" type	15,000 lb/hr max., working pressure 150 psi
61	2	Fuel Oil Burners	Best	Rotary cup, M.D.	50-600 lb/hr
62	1	L.O. Purifier (Turbine)	Sharples	Centrifugal, M.D.	300 gph, 2 hp
63	1	L.O. Purifier (Red. Gears)	Sharples	Centrifugal, M.D.	300 gph, 2 hp
64	2	F.O. Purifiers	Sharples	Centrifugal, M.D.	300 gph, 2 hp
65	2	Air Compressors	Ingersoll-Rand	2 stage, M.D.	10 ofla/min., discharge pressure 320 psi, 7½ hp
66	1	S.W. Evaporator	Originally installed	Coil	25 ton/day
67	1	Refrigeration Compressor	Originally installed	Freon 12, M.D.	10 hp
68	2	Aux. Generators	Engine—Superior	6 cyl., vert., S.A.	150 kw
			Generator—Elliott	Solid inj. 2 wire, 120 volt, d-c	
69	1	Emergency Generator	Originally installed	4 cyl., vert., S.A. 120 volt, d-c	10 kw

All necessary controls, telegraph, fire detecting and extinguishing systems, tanks, lifting gear, etc., will be included.

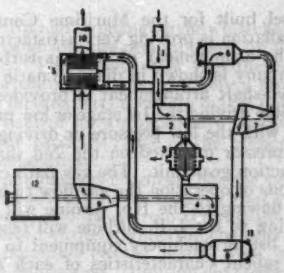
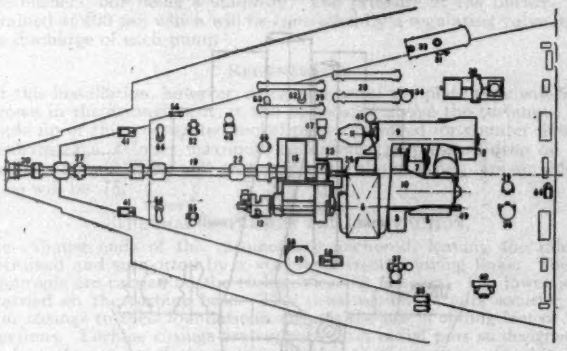
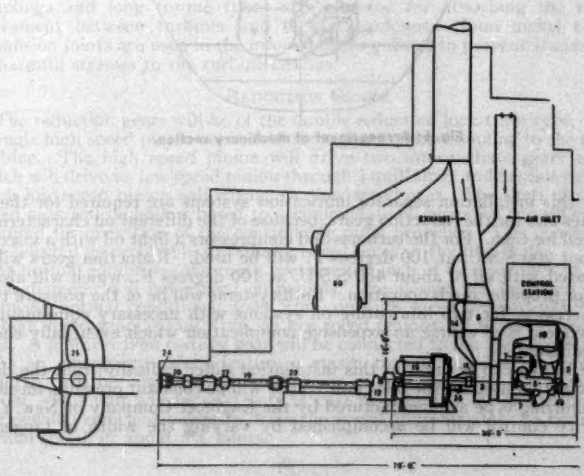


Fig. 3, right: Diagram arrangement of gas turbine. (1) Intake silencer and filter. (2) L. P. compressor. (3) Intercooler. (4) H. P. compressor. (5) Regenerator. (6) H. P. combustion chamber. (7) H. P. turbine. (8) L. P. combustion chamber. (9) L. P. turbine. (10) Exhaust to stack. (11) P. O. burner. (12) Reduction gear.

Fig. 4, below, top: Arrangement of machinery plan.
Fig. 5, below, bottom: Arrangement of machinery elevation.





vessel built for the Maritime Commission. My information is that this installation is proving very satisfactory.

The arrangement of the gas turbine unit to be furnished by the Elliott Company is shown in diagrammatic form in Figure 3. It will be noted that a two-shaft arrangement is provided, with the high pressure or controlling turbine driving the 1st stage or low pressure compressor mounted on one shaft, and with the low pressure or driving turbine and 2nd stage or high pressure compressor mounted on the 2nd shaft, which will also be connected to the reduction gear unit. The gas path can readily be traced. For various conditions of operation the gas flow to both turbines and the temperature of the gas flowing to the Hp. turbine will vary, while the temperature of the gas flowing to the L.P. turbine will remain approximately constant.

A list of machinery equipment to be installed in the vessel, together with the salient characteristics of each equipment item, is shown in the table herewith. The arrangement of the principal machinery items in the vessel is shown in Figures 4, 5 and 6. This arrangement has been adapted to the space allocated for machinery in the original design.

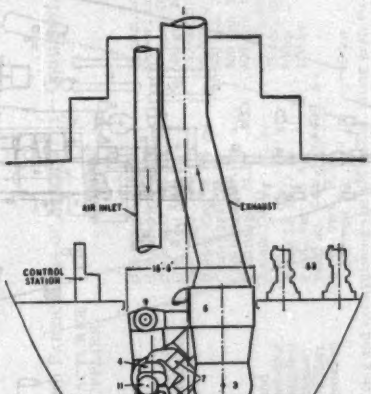


Fig. 6: Arrangement of machinery section.

For this installation separate lubrication systems are required for the gas turbines and for the reduction gears, because of the different oil characteristics required for each. For the turbines and compressors a light oil with a viscosity of about 200 S.S.U. at 100 degrees F. will be used. Reduction gears will be lubricated with oil of about 400 S.S.U. at 100 degrees F., which will also be used for propeller pitch operation. Both systems will be of the pressure type. The necessity for two lubricating oil systems with necessary equipment for each system is of course an expensive complication which eventually should be eliminated.

The burners to be used in this installation differ radically from the diesel type injection used for the pilot unit. They will be constant pressure, mechanical atomizing type as manufactured by the Engineer Company of New York. Capacity control will be accomplished by varying the width of tangential

slots in the whirl chamber at the burner tip and by changing the whirl chamber volume and the orifice opening. These variations are accomplished by means of a single moving rod within the burner. Test results show that the burner capacity with relation to the position of the rod is approximately a straight line for any given supply pressure. This fortunate relationship makes possible the design of a simple pneumatic type control for automatically positioning the rod and controlling combustion. The capacity range of the burner will be well above the requirements of the installation.

Two burners will be installed in each combustion chamber, one for normal operation and one stand-by. The installation will be so designed that either burner can be retracted into the cool zone, and away from the radiant heat of the chamber when not in use, or removed entirely while the chamber is in operation. Ignition will be by high potential electric spark from heavy electrodes, which will be used only when lighting off. Ignition electrodes will be arranged for retraction from the high temperature zone. The introduction and retraction of the electrodes will be controlled from the operating station.

There will be three rotary gear type motor driven pumps for supplying oil to the burners, one being a stand-by. The pressure at the burners will be maintained at 400 psi, which will be controlled by a regulating valve installed in the discharge of each pump.

Regenerator.

For this installation, however, a regenerator of the plate type will be used. As shown in the arrangement, it will be located above the turbines. It will be made up of thin corrugated nickel plates arranged for counter flow of air and exhaust gas. Under maximum conditions the pressure drop on the air side will be about .27 psi and on the gas side about .4 psi. Its air side effectiveness will be .75.

Thermal Expansion and Contraction.

The exhaust ends of the turbines are anchored, leaving the inlet ends unrestrained and supported by a system of freely moving links. The upper link journals are carried on the turbine bearing housing. The lower journals are carried on the turbine base. This construction permits securing of the turbine casings to their foundations without the use of sliding feet or flexible connections. Turbine casings are supported on radial pins so designed as to allow free movement of the casing relative to the rotors. Double flexible couplings and long torque tubes are provided for absorbing the relative movement between turbines and driven machines. Spun nickel toroidal expansion joints are used in the interconnecting ducts to prevent transmission of harmful stresses to the turbine casings.

Reduction Gears.

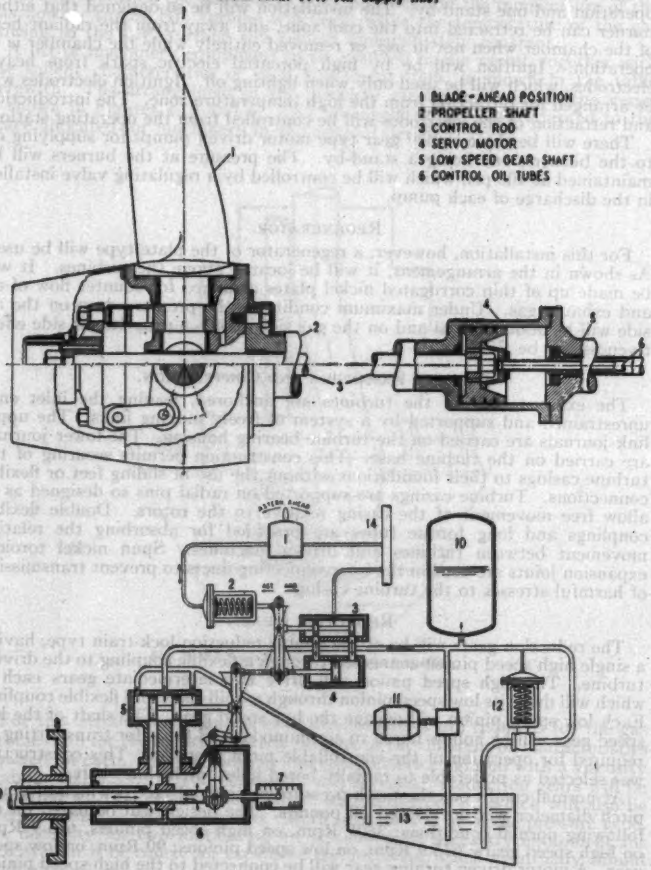
The reduction gears will be of the double reduction lock-train type, having a single high speed pinion connected through a flexible coupling to the driving turbine. The high speed pinion will drive two intermediate gears each of which will drive its low speed pinion through a quill shaft and flexible coupling. Each low speed pinion will engage the low speed gear. The shaft of the low speed gear will be hollow bored to accommodate oil pipes for transmitting oil required for operation of the controllable pitch propeller. This construction was selected as preferable to radially bored holes in the line shaft.

At normal conditions the tooth pressure per inch of face per inch of pinion pitch diameter will not exceed 60 pounds. The design will be based on the following normal conditions: 3609 Rpm. on high speed pinions, 675.9 Rpm. on high speed gears; 675.9 Rpm. on low speed pinions; 90 Rpm. on low speed gear. A motor driven turning gear will be connected to the high speed pinion.

As an emergency safety feature a steam turbine capable of developing 300 Hp. will be connected through gears and a clutch to the high speed pinion. This unit will be used only in case of failure of the gas turbine and will develop sufficient power to drive the vessel approximately 45 per cent of normal speed, or about 5½ knots.

Fig. 7, top: Controllable pitch propeller.

Fig. 8, bottom: Propeller pitch control. (1) Pitch controller. (2) Air control valve. (3) Air relay valve. (4) Air power piston. (5) Servo motor control valve. (6) Oil distributor head. (7) Pitch indicator. (8) Main thrust housing. (9) Servo motor supply tubes. (10) Hydraulic pressure tank. (11) Pump. (12) Bypass valve. (13) Oil drain tank. (14) Air supply line.



PROPELLER.

Since the gas turbine is non-reversible, the use of reduction gear drive necessitated the use of a controllable pitch propeller for backing and maneuvering operations. The installation consists of a 4-bladed propeller mounted on a hollow tailshaft and connected by 2 lengths of hollow lineshafting to the coupling of the reduction gear. The control mechanism will be designed to move the propeller blades from full ahead position to full astern position in 6 seconds when the vessel is proceeding ahead under normal power. A sectional view of the propeller is shown in Figure 7. Complete model basin tests of the propeller will be made.

The propeller blades will be hydraulically actuated and will be controlled by arms and linkage actuated by a control rod extending through the hollow shafting from the propeller hub to a servomotor piston within its cylinder in the foremost length of lineshafting. Two concentric steel tubes pass through the hollow shaft of the low speed element of the reduction gear for transmission of oil from a distributor head mounted forward of the reduction gears, to the servomotor cylinder. The inner of these tubes will be attached to the servomotor piston, and will supply oil to the after side of the piston, and will move with the piston. The follow-up mechanism on the oil distributor head will be actuated by the motion of the inner tube. Oil to the distributor head will be controlled by the position of a 4-way hydraulic control valve which in turn will be controlled by the position of an air operated pilot valve. The pilot valve will be controlled from a pneumatic control system with operating valve at the central control station, as shown diagrammatically in Figure 8.

AUXILIARY BOILER.

A 150 pound auxiliary steam system will be provided which will be supplied by a boiler of the two-drum type having a maximum capacity of 15,000 pounds of steam per hour. This system will supply steam for operating deck winches, anchor windlass, steam operated auxiliaries in the engine room, and for heating purposes. The boiler will be fitted with 2 oil burners of the rotary cup type, in the design of which will be incorporated automatic combustion and flame failure controls. There will also be fitted a small burner with automatic ignition for low capacity use.

TURBINE CONTROLS.

The design of the speed governor will be based on the temperature-revolution characteristics of the turbines and on the capacity-revolution characteristics of the compressors, both of which are basically straight line functions. The control will be sensitive to variations in speed of the driving turbine. It will operate to vary the fuel supply to the high pressure combustion chamber with resulting variation of gas temperature to the controlling turbine, and consequent variation of revolutions of that turbine and of the low pressure compressor which is directly connected to it. The resultant effect of speed variations will therefore be variations in the supply of air to the system and consequent variations of gas flow to the turbines.

Since variation of air flow in the system causes variation of the gas temperature entering the low pressure turbine, and since the best efficiency can be obtained by maintaining maximum temperature at the low pressure turbine inlet, a thermostatic temperature controller for this purpose will be provided. This controller will vary the fuel entering the low pressure combustion chamber as required.

Both combustion chamber discharges will be provided with controls which will function to shut off the fuel supply in the event of temperatures exceeding predetermined limits. There will also be a temperature control on the high pressure combustion chamber which will be superimposed on the speed governor and which will reduce the fuel supply to that chamber when the gas temperature approaches a predetermined limit. The purpose of this control is to prevent the operator from increasing the load on the plant too rapidly.

An increase in speed over the governor setting will cause a decrease in fuel flow to a set minimum value. If at that setting the speed still increases, the overspeed control or trip will operate to cut off all fuel flow and open a dump valve releasing air from the high pressure compressor discharge to the atmosphere, which will stop both turbines.

The underspeed control is not a separate control but a feature of the oil burner design which prevents complete shutoff of the fuel. Its function is to prevent stalling of the unit under maneuvering conditions.

Protection against flame failure will be accomplished by means of temperature sensitive elements fitted in each combustion chamber which will operate to shut off the fuel supply in event of flame failure.

It is of interest to note that only one manual speed governor setting is required for the control of the entire gas turbine unit, once it is in running condition. Manual emergency operation of all controls will be provided.

Complete instrumentation and control operation will be centered at the operating station.

STEAM JET REFRIGERATION FOR MARINE PURPOSES.

This very elementary discussion of steam jet refrigeration or cooling is reprinted from "The Marine Engineer" of November 1945 to acquaint Naval Engineers with this comparatively new type of cooling for sea going installations. Several trial installations are being made currently on Naval Vessels which will join the Fleet in the next year. The article is an abstract of a paper presented to the "Institute of Marine Engineers" by W. Sampson.

The term "refrigeration" covers cooling or freezing, and it should be stated that the field for steam jet refrigeration is mainly in the cooling range, for although there have been plants built and operated at temperatures well below the freezing point, using brine as the heat-carrying medium, these plants are designed as series jet machines and are of a low efficiency.

Confining steam jet refrigeration units to the cooling field means that fresh water can be the heat carrier; this brings in train many advantages. To cool from atmospheric temperatures down to freezing temperatures there is a big heat load to be handled, and because of the advantages of single-stage jet refrigeration over other forms of refrigeration in its own field, it is becoming more and more recognized.

If the heat-carrying medium, say water, is introduced into a chamber which is maintained at a low absolute pressure (high vacuum) this water will boil. The latent heat required for evaporation of part of this water comes from the total mass of water, resulting in a lowering of the sensible heat of the mass with a corresponding drop in temperature of the total mass.

The required vacuum is obtained by means of a thermo-compressor (steam jet) which draws the vapor from the chamber, usually called the flash chamber. The vapor is compressed to a condenser where a vacuum is maintained, corresponding to the temperature of the available sea water, and thus the heat brought in by the water to the flash chamber is thrown overboard in the condenser discharge.

The main essentials of a steam jet plant as shown in Figure 1 are: (1) vacuum chamber called the flash chamber; (2) thermo-compressor; (3) condenser, with air ejectors; (4) pumps circulating chilled water to the heat receiving apparatus, where heat is taken up by the chilled water and taken down to the refrigerator. These heat receiving plants may be air coolers, as in air conditioning plants, water coolers, or liquid CO_2 coolers; (5) condensate pump returning condensed steam to the feed system.

The flash chamber is merely a mild-steel welded vessel into which the warm water is sprayed. Vapor is drawn off the upper half of the chamber by the steam jet thermo-compressors. Where more than one thermo-compressor is used the flash chamber is divided into pressure-tight compartments, one for each thermo-compressor. This enables one thermo-compressor to be shut

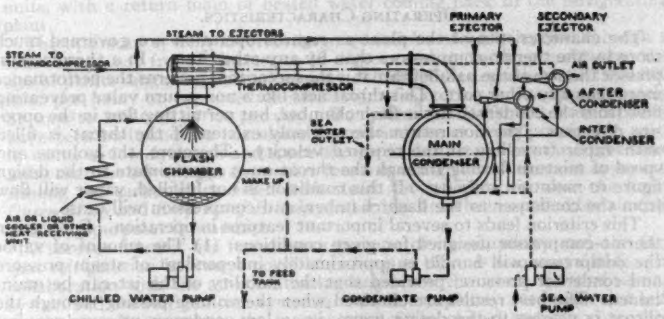


Fig. 1—Typical diagram of steam jet refrigeration plant

down, without vapor blowing back from the condenser, by merely shutting the appropriate steam valve. The water level is carried near the bottom of the flash chamber, as most of the evaporation takes place from the surface of the sprays of warm water. This means that, in the case of multiple compressor plants, the chilled water pipes from the compartments should be separate and long enough to allow for depression of the water in the leg from a compartment which is at condenser pressure.

Thermo-compressor Action.

The thermo compressors, Figure 2, have one or more high-velocity steam jets expanding the actuating steam to flash chamber vacuum. These jets entrain and accelerate the refrigerant vapor from the flash chamber in a convergent tube. The mixture of steam and vapor is projected through a parallel throat section, and then is decelerated in a divergent section where the velocity energy of the mixture is converted into pressure energy. The vapor head may be of cast or fabricated construction. The combining tube is usually of cast construction in iron or bronze, and the steam chest is of cast-iron, bronze or steel, depending on the steam conditions. The condenser is of the surface type, and is standard marine design. The steam jet air pump and the centrifugal pumps are all standard equipment.

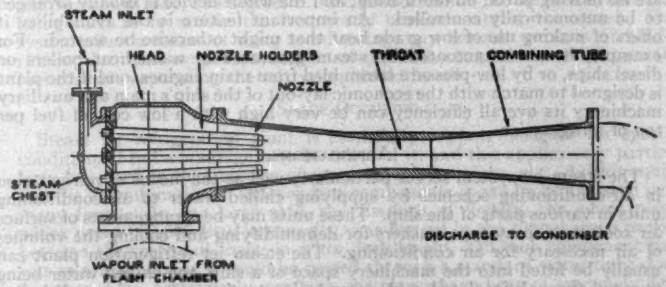


Fig. 2—Sectional view of thermo-compressor

Operating Characteristics.

The characteristics of the plant as regards operation are governed much more by the thermo-compressors than by any other item. In a thermo-compressor the size of the parallel throat is the factor that governs the performance more than any other part. This throat acts like a non-return valve preventing flow from the condenser to the flash chamber, but permitting flow in the opposite direction. The non-return effect is only existent if the throat is filled with vapor travelling at the required velocity. Therefore, the volume and speed of mixture flowing through the throat must approximate to the design figure to maintain stability. If this condition is not fulfilled, vapor will flow from the condenser to the flash chamber, and compression will cease.

This criterion leads to several important features in operation. For a given thermo-compressor designed for given conditions: (1) The amount of vapor the compressor will handle is approximately independent of steam pressure and condenser pressure, provided that the stability of the jet can be maintained. The best results are obtained when the volume passing through the throat is nearest to the design figure, i.e. a low condenser pressure requires a low steam pressure. (2) The amount of vapor the thermo-compressor will handle is dependent on the suction pressure. (3) The absolute steam pressure at the nozzles required to maintain stability is proportional to the absolute pressure in the condenser. This is fairly obvious as the volume per pound of the mixture at the throat is proportional to absolute condenser pressure, and the pounds of mixture passing are approximately proportional to the absolute steam pressure at the steam nozzles.

The above remarks cover the performance of a thermo-compressor designed for a given set of conditions. To appreciate how the designed steam consumption of a thermo-compressor varies, it is best to consider the thermo-dynamics of the design. Steam is being expanded through a nozzle, and heat is being converted into velocity. The velocity attained by the steam depends on the available adiabatic heat drop between steam conditions and flash chamber vacuum. This steam entrains the vapor, and is compressed to the condenser pressure. The velocity the mixture must attain to give the necessary compression is again dependent on the heat drop from the condenser to the flash chamber. Therefore, the ratio of actuating steam required to vapor compressed depends on the velocity of the steam after expanding to flash chamber vacuum as compared with the velocity of the mixture needed to give the necessary compression ratio. These velocities depend in each case on the adiabatic heat drop. Thus, it will be seen that the ratio of steam used to vapor compressed decreases with increase in the steam pressure, and with decrease in compression ratio between the flash chamber and condenser.

Advantages of Steam Jet.

The advantages are simplicity, as steam only is used; absence of mechanical parts; noiselessness; no attention on the part of the operator is needed as there are no moving parts; no lubrication; and the whole device is usually arranged to be automatically controlled. An important feature is the availability it offers of making use of low grade heat that might otherwise be wasted. For example, it can be actuated by steam generated in waste-heat boilers on diesel ships, or by low-pressure steam bled from main engines, and if the plant is designed to match with the economic lay-out of the ship's main and auxiliary machinery its overall efficiency can be very high with a low cost of fuel per ton of refrigeration.

Applications.

The steam jet system serves particularly well as the means of refrigeration in air-conditioning schemes by supplying chilled water to air-conditioning units in various parts of the ship. These units may be arrangements of surface air coolers or spray-type washers for dehumidifying and cooling the volumes of air necessary for air conditioning. The steam jet refrigeration plant can usually be fitted into the machinery space of a ship, the chilled water being pumped through insulated cold water mains to the various air conditioning

units, with a return main of heated water coming back to the refrigerating plant.

The optimum system involves large quantities of water cooled through a small range, say from 7 degrees to 10 degrees and up to the present time the size of plants ranges between a capacity of 250,000 Btu. per hour to 2,000,000 Btu. per hour, or expressed in terms of refrigerating units approximately 21 tons to 160 tons of ice per day. But plants of 10,000,000 Btu. capacity have been built and are postulated for marine use as being practicable units in terms of low first cost and space occupied. In fact, it is in the larger-sized units that the advantages over other forms of refrigeration are most apparent. Among the earlier ships fitted with steam jet refrigeration were the liners *Queen Mary* and *Empress of Britain*, and it has since been applied for air conditioning purposes in warships.

Wherever fitted it has found favor because of its simplicity and the fact that refrigeration is always possible as long as there is steam available. In collaboration with the air-conditioning engineers, the whole air-conditioning system and the refrigeration system are made completely automatic and self-regulating. For instance, the temperature of the chilled water is controlled within plus or minus 1.5 degrees F. over widely-varying air-conditioning loads by simple thermostatic governors regulating the steam to the thermo-compressors.

There is at present an installation being installed of 370,000 Btu. capacity in conjunction with the air conditioning of all living quarters in a tanker. Here advantage has been taken of the features of the steam jet refrigeration plant to use the same equipment for delivering heated water to the air conditioning units by the fitting of a calorifier in the water main, thus using the same units for winter heating. Advantage has also been taken to fit drinking-water coolers at various points, using the chilled water from the refrigerating plant as a coolant.

Combination Plants.

Assuming a modern ship had an air-conditioning load, a liquid CO_2 cooling load and an evaporator load, then a very simple and economical combination can be made. There is a flash chamber divided into three parts. In one section salt water is cooled and partially vaporized, which in turn cools fresh water which is the refrigerant for the air-conditioning load. In another chamber salt water is cooled and vaporized and in turn cools liquid CO_2. Each of these two chambers would, of course, have their independent thermo-compressor. These would compress to the condenser in which there would be carried a vacuum corresponding to the sea temperature, which with a sea temperature of say 85 degrees F. would be 28.0 inches. The third section of the flash chamber would be made common with the condenser through a pipe duct. Therefore the vacuum in the third section of the flash chamber would be also 28 inches, i.e. approximately 100 degrees F. Into this chamber the necessary salt water to be evaporated would flow through sprays at an entering temperature of, say, 150 degrees F., the quantities and the amount evaporated being such as to satisfy the fresh-water requirements of the vessel. This system offers a means of using to advantage any hot salt water available for running into this evaporator section of the flash chamber, in which case the fuel cost of evaporating the salt water would be nil.

Space Occupied.

Steam jet refrigeration plant is essentially a plant designed for specific conditions if full advantage is to be taken of it, and this applies very particularly in regard to the question of space occupied, weight, etc. As this equipment is a combination of independent components it can be arranged in very many forms. In one large ship, the *Empress of Britain*, the whole plant, which was 1,000,000 Btu. in capacity, was housed in the tunnel space packed between the spare tail shafts in a low deck height, and each component part was so arranged that it could be passed through a standard water-tight door. In other ships, particularly warships, the plants have been of extremely light

weight, and so made that each part could be passed through armored hatches. All have been arranged with the condensers, flash chambers, etc., in various relative positions, sometimes well spread, sometimes at different angles according to the floor space available and the deck heights.

In two ships now building the vacuum refrigeration plants will be spaced adjacent to the boilers and arranged vertically between the boilers and the ship's side, occupying very little width and length but greater height. In an installation now being fitted in a tanker for air-conditioning purposes, the whole equipment has been arranged on a stringer at the after end of the engine room. Plants can also be arranged on different decks and levels. For instance, the pumps may be on the engine-room floor, the flash chamber and condenser on the deck above, or even higher.

WEIGHTS OF STEAM JET REFRIGERATION PLANTS.

Output of Plant Tons refrigeration	Weight of Plant Tons	Ratio Tons/Ton refrig.
40	8	0.2
80	11½	0.144
240	25½	0.106

ON THE MATHEMATICS OF COMMITTEES, BOARDS, AND PANELS.

This is a humorous article prepared by Bruce S. Olds of the Arthur D. Little, Corp., Cambridge, Massachusetts. It appeared originally in the "Scientific Monthly" of August 1946 and is reprinted from that publication.

The present is considered to be a most appropriate time to study analytically, with a view toward improving, the efficiency of functioning of committees, boards, and panels in general. Two reasons supporting this stand are:

(1) The prosecution of scientific work during World War II was largely in the hands of committees, boards, and panels (*miserabile horribileque dictu!*).

(2) The security regulations still in operation, such as the Espionage Act, prevent the publishing of interesting treatises and force technical journals to accept almost anything.

The three items under consideration can be defined briefly in the following manner:

(1) *Committee.* A body of persons appointed to consider, investigate, or take action upon, and usually to report concerning some matter.

(2) *Board.* A council convened for business. (There is absolutely no authenticity to the definition "long, narrow, and wooden" sometimes applied.)

(3) *Panel.* A list or group of persons appointed for some services.

It is immediately apparent from these definitions that committees, boards, and panels are similar in that all are groups of one or more persons formed *to accomplish work.* Thus they will be treated simultaneously in this paper.

There are numerous methods of expressing mathematically the objective of committees, boards, and panels: namely, to perform work. The most commonly applied formulae are the following:

(1) The utilization of the familiar time, force, and distance relationship where the object is to maximize the expression

$$\text{ComBulPac} = \frac{fd}{t}$$

in which ComBulPac = code name for power output of committees, boards, and panels, f = force, d = distance, and t = time.

This equation was the origin in 1812 of the expression "a powerful committee."

(2) A second and very appropriate method is the use of the gas law as a basis for the work expression. Here the attempt is made to maximize final minus initial gas volume:

$$\text{ComBulPac} = \int_{v_1}^{v_2} p\,dv$$

in which p = pressure and v = volume.

It is held by experts that Gay-Lussac first evolved from this relationship the phrase "a high-pressure committee." On the other hand, there is no foundation for the rumor that the current slang phrase, "the committee is cooking with gas," was derived from this equation.

(3) Another method which is very useful under certain circumstances is

$$\text{ComBulPac} = Le$$

in which $L = Fr$, F = force, r = radius, and e = angle.

This, the reader will recognize, is the well-known "revolving committee.'

(4) A fourth method of setting down the work expression is becoming very common since the importance of air power has been realized:

$$\text{ComBulPac} = M\,(V) - C_d \frac{\rho A V^2}{2}$$

in which M = mass flow, V = velocity, C_d = drag coefficient, ρ = density, and A = area.

Recognizable immediately, here is the "committee with drag." (Drag is high for certain simple bodies.)

(5) A fifth method of calculating output has been evolved during World War II. This is to express the output for the widely used "joint" committee:

$$M = \frac{\gamma \Sigma Y^2}{y}$$

in which M = resistance to moment of a group of rivets in a riveted joint, γ = total allowable stress per rivet (or committee member), ΣY^2 = sum of squares of distances from center of gravity of the group of rivets, and y = distance of outermost rivet from center of gravity of the group.

The reader will note that, although elastic bending under stress of great magnitude occurs, no work is accomplished by the joint committee.

In order to determine which of the five equations best expressed the output obtained, the performance since December 1941 of a large number of committees was analyzed. Very poor correlations were found between the actual and theoretical calculated work outputs. These results are plotted in Figure 1 for all five methods of calculation.

This poor correlation showing the actual output of wise decisions, good reports, and constructive accomplishments per year, always far below theoretical, led to the decision that a much more thorough analysis than had been made heretofore would have to be undertaken. The importance of this study to the war effort was such that an overriding priority was assigned. (By a fortunate error, the file clerk placed this problem on the top of the growing pile of over-riding priority projects so that it took an over-riding precedence and was completed only seven months behind schedule.)

Analysis. The most logical mathematical approach to the problem of calculating accurately the output of committees appeared to be the application of the method of multiple correlation. This tedious procedure was therefore used. Certain machines, such as the one at Harvard College, were of invaluable assistance in carrying out the calculations.

It was decided that the work output must be such that

$$W_o = E_2 W_i$$

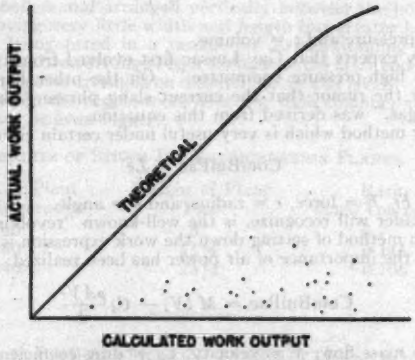


FIG. 1. WORK OF COMMITTEES

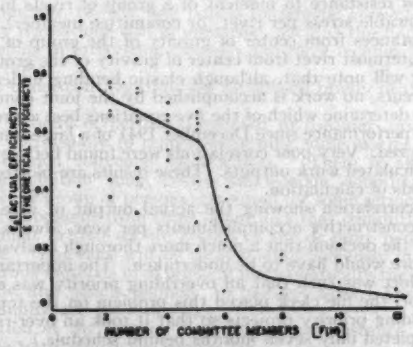


FIG. 2. EFFICIENCY VS. NUMBERS

where W_a = actual work, W_t = theoretical work, and E_2 = actual efficiency of the committee.

Since it was believed that W_t could be calculated quite accurately, it became obvious that a study of the factors affecting the efficiency of operation of committees, boards, and panels leading toward an accurate calculation of the efficiency, E_2, was the key to the problem.

Therefore, as a first step an equation was written in the following form:

$$E_1 = f(n)\, f(i)\, f(c)\, f(hs)\, f(m)\, E \qquad (1)$$

in which E = theoretical committee efficiency, E_1 = calculated committee efficiency, $f(n)$ = function of number of committee members, $f(i)$ = function of intelligence of committee members, $f(c)$ = function of type of committee chairman, $f(hs)$ = function of type of hecklers and saboteurs on committee, and $f(m)$ = function of the miscellaneous element; and where the calculated committee efficiency is expressed as a function of various parameters times the theoretical efficiency. The object is to arrive at the proper values of the various parameters so that

$$E_1 = E_2$$

where E_1 is the calculated and E_2 the actual efficiency.

A thorough study of the parameters in equation (1) will now be undertaken.

Many examples show that the number of men on a committee, $f(n)$, affects very materially the work accomplished. These data are plotted in Figure 2. It is apparent that it is best in many cases to have the membership limited to one, and membership of over five is usually fatal.

One striking thing to note about Figure 2 is the large scatter of points for any one value of the number of committee members. This was interpreted to mean that other parameters were affecting the data, such as the intelligence, etc., of the individual members. That this is actually the case will now be shown.

A large number of observations have been made of the effect on output of the intelligence, $f(i)$, of individual committee members. This matter of abilities of personnel on committees can usually be reduced to the following simple terms of division:

(1) Working-level personnel who know the details of the subject under consideration.

(2) Policy-level personnel who know no details of the subject under consideration.

Thus in forming a committee, one is confronted with the decision as to whether he wishes the membership to be composed of working- or policy-level personnel. The war has presented a unique opportunity for arriving at a proper answer to this question. It so happens that the rank of a military man is directly related to the degree to which he is a policy man, thus allowing an absolute measure to be applied. Pursuing this promising lead, a remarkably fine set of data was collected, thus allowing the establishment, as follows, of perhaps the most fundamental law discovered in this paper:

$$I = \frac{E}{R} \text{ (Old's Law)}$$

in which I = intelligence in any given subject, R = rank of individual, and E = a constant of very small magnitude of the order of $1/c$, where c is the velocity of light. This indicates that E has the dimensions of a wave "slowness."

As can readily be seen from Figure 3, there is little deviation from the law except at the extremities. (In this regard it maintains its similarity to Ohm's Law, $I = E/R$.) It follows that it is a simple matter to deduce the fact that if one desires to form a committee with high efficiency of work output, it is essential to select working-level personnel. Data supporting this statement are plotted in Figure 4. As would be expected, there are individual exceptions which appear as points off the curve, and those will now be analyzed.

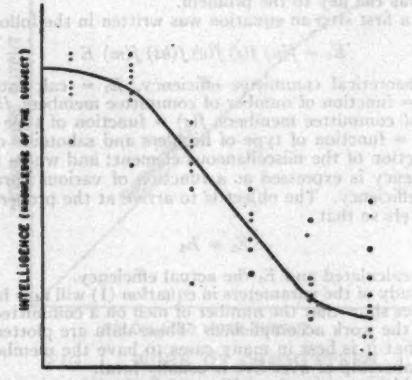


FIG. 3. INTELLIGENCE VS. RANK

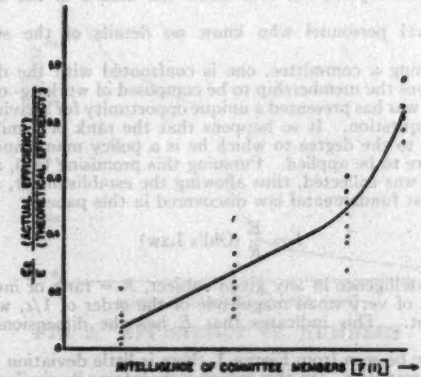


FIG. 4. EFFICIENCY VS. INTELLIGENCE

One factor which affects strongly the ability of even an intelligent committee to do useful work is the capability of the chairman of the committee ($f(c)$). Let us type the characteristics of committee chairmen in the following manner:

(*A*) A really capable man who knows the subject, has well-prepared agenda distributed before the meeting, skillfully keeps people on the subject (but not to the extent that he does not allow both sides of questions to be thoroughly exposed), requires that action be taken on agreements reached, follows up on such actions, makes efficient use of the method of task assignment to subcommittees, provides for periodic needling of the committee by outside experts, requires written comment on reports circulated to members, insists on a minimum of six committee meetings per year, and employs an efficient secretariat.

(*B*) A man similar to Type *A* in every respect except that he is too nice a fellow to interrupt ramblers, particularly if they happen to occupy a higher position or are older than he.

(*C*) The man who is one of the leaders in the field under consideration but who uses the committee merely as an instrument to second his ideas. In cases where questionable ideas have to be forced through, he lines up his votes beforehand or rudely interrupts all opposition.

(*D*) The chairman who is obviously too important for the committee. He sends a deputy to run the meetings with instructions to hold off on any really important decisions until he can find time to attend and whip things into shape.

(*E*) The chairman who opens the meeting by frowning slightly and saying: "Now, er, ah, let's see. Heh, hrrmphh, ah. Who called this meeting? I mean, what are we here for today?"

The strong effect of the type of chairman, $f(c)$, on the efficiency of a committee is shown in Figure 5.

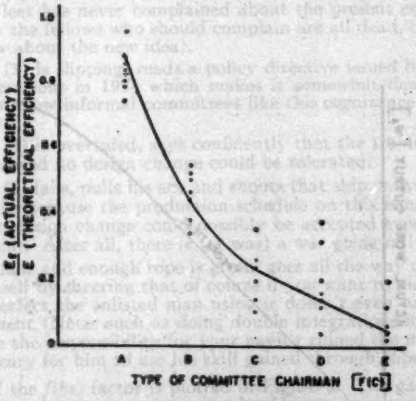


FIG. 5. TYPE OF CHAIRMAN

There are some conditions which can ruin the efficiency of a committee even though it may have intelligent members and a competent chairman. The most serious of these is the heckler-saboteur function, $f(hs)$. The main types of heckler-saboteurs, many of which are well known, are as follows:

(*A*) The normal man. (All *Homo sapiens* have some faults.)

(*B*) The jolly fellow who is always 22 minutes late and then holds up proceedings 7 more minutes telling (off the record) his latest joke. Although he never does any committee work between meetings, he is such a good egg he never fails to get reappointed.

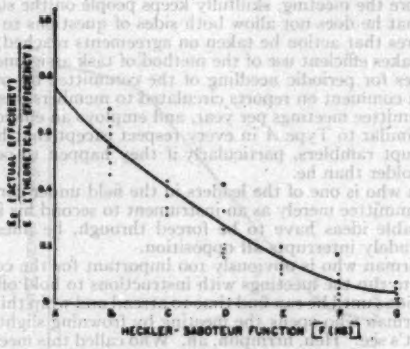


FIG. 6. TYPE OF SABOTEUR

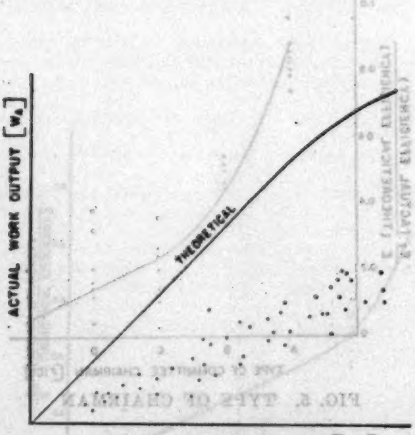


FIG. 7. WORK RECALCULATED

(*C*) The man with an elephantine memory to whom all ideas are old and w o can still quote all the reasons used to turn any one of them down 11 or 12 years ago. No chance for viewing anything in a new light is given—the mere fact that he had heard of it before is sufficient reason in his mind to vote against any idea.

(*D*) The man who is against initiating any work, because, since there is a shortage of scientific manpower, any new project undertaken is bound to interfere with the progress of all existing projects (particularly two of his pet programs).

(*E*) The poor fellow sent to represent his boss who has instructed him in such a manner that all he can do is sit there and say, "I don't know," or "I have no authority to speak for my agency."

(*F*) The policy man who is afraid the rest of the committee is trying to take away some of his power and authority. Thus, he views each question not from the standpoint of whether it is the best thing to do, but whether the answer given might possibly be misinterpreted by anyone as permission for someone to infringe upon his cognizant empire.

(*G*) The man who is on the defensive. He suspects the committee has been formed just to change (Note: for the better) his method of doing something. His actions are almost bound to hew to the following pattern:

(1) Announces that his office carefully considered this idea, which is really not basically a new one, about a year ago, and decided to turn it down in favor of the design now in use.

(2) When (1) is attacked successfully, states rather emphatically that this new design is just the idea of some long-haired, impractical professor who doesn't understand the wear and tear on this gear out in operation.

(3) When (2) is successfully countered by the opposition, says proudly that the Fleet has never complained about the present equipment (but fails to say the fellows who should complain are all dead, or too busy, or don't know about the new idea).

(4) When (3) is slipping, reads a policy directive issued by the Chief of Naval Operations in 1924 which makes it somewhat doubtful whether regulations allow informal committees like this cognizance to recommend a change.

(5) When (4) is overruled, says confidently that the training program is so far advanced no design change could be tolerated.

(6) When (5) fails, pulls his ace and shouts that ship deliveries are being held up now because the production schedule on this equipment is way off, and no design change could possibly be accepted even if it were an improvement. After all, there is (or was) a war going on!

(7) If (6) fails and enough rope is given, goes all the way out and openly hangs himself by sneering that of course if you want to make this equipment so perfect the enlisted man using it doesn't even have to exercise any judgment (Note: such as doing double integrations in his head), he won't take the responsibility for your having ruined the man by making it unnecessary for him to use his skill gained through supertraining.

The effect of the $f(hs)$ factor is plotted in Figure 6. The gain that can be made by selecting A type committee members is very apparent.

Finally, there are several factors that can best be termed "miscellaneous" which upon occasion seriously affect committee efficiency. Among these are such things as the temperature of the conference room, the degree of comfort of the meeting chairs, the size of lunch served, the amount of time wasted during the meeting in arranging for train and hotel reservations, and whether a quorum of committee members happened to get together the night before the meeting and settle all the expected controversies of the morrow in the bar. The latter practice is recommended by many competent committee chairmen but requires more research before definite conclusions can be reached.

From the nature of the miscellaneous factor, $f(m)$, it is apparent that this parameter must be estimated for each special case.

Results. Using a slightly modified standard form of multiple correlation calculation to evolve the proper parameters for the several variables a variance of 0.3 was obtained for correlation between calculated and actual committee efficiencies. Snedecor's test of significance showed this multiple correlation to be almost significant.

Employing corrected calculated efficiencies, the actual (W_a) and calculated (E_3W_4) committee work outputs have been replotted in Figure 7. It will be noted that, while a distinct improvement has been made over the original data plotted in Figure 1, the actual work output of committees is still disappointingly far below theoretical.

Conclusions. The lack of correlation achieved in this paper is regretted. It may be that the choice of parameters was completely unsound. One point which particularly baffles the author is the peaking of the efficiency of output of a committee versus number of committee members (Figure 2) at seven-tenths of a person. Obviously one must conclude that either further research is required or that people are no damned good.

BOOK REVIEW.

SECRET MISSIONS, By Captain Ellis M. Zacharias, U. S. N., Published By G. P. Putnam's Sons, New York, New York.

This book is an account of the author's experiences and impressions as a Japanese language student and his frequent duties in the Office of Naval Intelligence preceding and during World War II. He gives an interesting insight into the character and thinking of the leading Japanese Naval Wartime figures and tells of the work of our own intelligence activities in breaking spy rings, preventing sabotage, and arresting of Axis fifth columnists. He discusses the Pearl Harbor attack and the intelligence behind it, including the highly controversial "Winds" messages. In his closing chapters he tells of the psychological warfare against the Japanese and the exploiting of their internal weaknesses to destroy their will to continue the war.

LANGE'S HANDBOOK OF CHEMISTRY—SIXTH EDITION. Published by Handbook Publishers, Inc., Sandusky, Ohio.

This sixth edition of the familiar *Handbook of Chemistry* is completely up to date with revised tables and new material such as a section on *Modern Concepts of Matter*. The usual sections covering physical and chemical properties of the Elements, Inorganic Compounds, and Organic Compounds, are included along with sections on Analytical Chemistry, Physical Chemistry, Industrial Chemistry and Mathematical Tables. To facilitate use of the Handbook a very complete index is provided.

WHITE HOUSE PHYSICIAN, By Vice Admiral Ross T. McIntire (M. C.), U. S. N. Published by G. P. Putnam's Sons, New York, New York.

This frank and intimate account of twelve years of close association with Franklin Roosevelt relates much of reasoning

and motives behind the historic decisions which the President had to make during his twelve years in office. Admiral McIntire accompanied the President on every trip at home and abroad and not only did he see and hear the people and the men who made history but he was *told* by the President himself of the events of the day. Particularly interesting are the President's consideration of Communism, and Russia's post-war course and conduct, his opinion of Chiang Kai-shek and the future of China, his reasons for approving atomic bomb research, and what he thought of strikes.

ASSOCIATION NOTES.

Society Officers and Council.

The following Officers and Council were elected for 1947:

President:

Rear Admiral C. W. Styer, U. S. N.

Secretary-Treasurer:

Captain F. W. Walton, U. S. N.

Members of Council:

Rear Admiral N. L. Rawlings, U. S. N.
Captain A. G. Mumma, U. S. N.
Captain K. K. Cowart, U. S. C. G.
Commander W. H. Spowers, Jr., U. S. N. R.
Mr. C. G. Cooper.
Mr. F. C. MacKrell.*

Annual Banquet.

The Annual Banquet of the Society will be held at the Statler Hotel, Washington, D. C., on Friday, 18 April 1947. Announcement and Reservation forms are being mailed to all members.

Society Scholarship.

The Society takes pleasure in announcing the establishment of scholarship arrangements with Webb Institute, New York.

These arrangements consist of total expenses of candidates who may qualify for entrance.

A brief of entrance requirements follows:

a. Be a dependent of member of the Society.
b. Be between 16 and 21 years of age. (Veterans may be excepted.)
c. Be an American citizen.
d. Be unmarried.
e. Be acceptable to the Institute (demonstrated by competitive written and oral examinations.)

*Appointed by the President to fill unexpired term of Mr. Post.

f. Have at least 15 credits from an accredited high school or equivalent, including:

1	Physics	½	Plane Trigonometry
4	English	2	Foreign Language
1½	Algebra	2	History (1 year of
1	Plane Geometry		which must be
½	Solid Geometry		American History)

g. Entrance examinations covering:

English
Physics
Algebra
Plane and Solid Geometry
Plane Trigonometry

The candidate must pass a physical examination showing him to be free from contagious diseases or deformities preventing full participation in school activities.

Full particulars will be contained in a circular addressed to all members; however, interested members may address inquiries to the Secretary-Treasurer.

Society Incorporation.

The Society has been incorporated under the laws of the District of Columbia as of 3 October, 1946.

Society Lapel Button.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a

very fine, dignified insignia. It is one-half inch in diameter. The oak leaves and lettering are red on a gold back-ground. It is available to all members at fifty cents (50c) each.

MEMBERSHIP.

Attention of all non-member officers is especially invited at this time to the advantage and benefit to be attained by membership in this Society. The Society was organized in 1888—for the sole purpose of the advancement of Engineering. Its JOURNAL, published quarterly, in February, May, August and November, and completing its 58th year of continuous publication, has attained a foremost rank in its field and is a recognized authority throughout the engineering world. It is used generally by Naval officers and civilians alike as a reliable reference book. The Bureau of Ships considers the JOURNAL of such value that it makes it available for the libraries of all Naval vessels and shore activities of any size. The presence of the JOURNAL in your personal library cannot fail to be of great benefit to you, often making available information of great value which can be obtained from no other source.

It is particularly suggested that officers of the Naval Reserve, who are returning to civil life, should arrange for membership prior to detachment from active duty.

Annual dues are $5.00, which are probably lower than those of any other technical Society. There is no initiation fee, nor any extra charge for the JOURNAL. This extremely low cost is possible only because the Society is bona fide operated solely for your benefit without any pecuniary profit to any one.

An application blank generally will be found at the back of each copy of the JOURNAL. However, if none is found, application by letter will be accepted.

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MEMBERSHIP ADDITIONS.

The following have joined the Society since the publication of the November, 1946, JOURNAL. Addresses of members on shipboard not given:

NAVAL.

André, Rufus Luke, Ensign, U. S. N. R., Research Engineer, Cutter Laboratories.
Mail 1767 Euclid Ave., Apt. 3, Berkeley, Calif.

Antolic, Joseph J., Lieut., j.g., U. S. N. R., S. S. *Marne*,
Room 2747 Graybar Building, New York, N. Y.

Barnes, R. M., Captain, U. S. N., Pearl Harbor Naval Shipyard,
Navy 129, care FPO San Francisco, Calif.

Bilderback, Arthur H., Lieut., U. S. N.,
1220½ East First St., Long Beach, Calif.

Booker, Thomas E., Ensign, U. S. N.,
821 E. Commerce St., Mexia, Texas.

Boyd, Earl I., Ensign, U. S. N., BOQ 228, Room 338,
U. S. Naval Station, Treasure Island, San Francisco, Calif.

Branan, Virgil C., Lieut. Commander, U. S. N. R., Port Engineer,
Strachen Shipper Co., Houston, Texas.
Mail 230 Ashburn St., Houston 12, Texas.

Brusstar, Delano, Commander, U. S. C. G. R.,
202 Park Ave., Apt. 610, New York 22, N. Y.

Candler, Lane, Ensign, U. S. N.,
Naval Officers Electronics Material School,
Treasure Island, San Francisco, Calif.

Cater, Charles J., Captain, U. S. N.

Chambers, James P., First Lieut., U. S. M. C. R.,
129 Myrtle St., Boston, Mass.

Cos, Grant A., Ensign, U. S. N.

Cosden, Thomas E., Lieut. Commander, U. S. N. R.,
4534 Brown St., Union City, N. J.

Duquette, Herbert E., Jr., Lieut., j.g., U. S. N.

Dwyer, John C., Ensign, U. S. N.,
6 Parkview Place, Baldwin, L. I., N. Y.

Efird, Terril A., Lieut., U. S. N.,
23 Morse Ave., Groton, Conn.

Ephror, G. O., Lieut., j.g., U. S. N.,
120 Carrollton Court, Norfolk, Va.

Foster, Stuart A., Staff Supervisor, Mechanical Installation Division, Newport News Shipbuilding & Dry Dock Co.
Mail 310 Piez Ave., Hilton Valley, Va.

Frana, B. T., Lieut Commander, U. S. N.
10 Dogwood St., Homoja Village, Naval Academy, Annapolis, Md.

Goddard, H. R., Jr., Lieut., U. S. N. R.,
3475 Valemont St., Point Loma, San Diego 6, Calif.

Graham, John G., Ensign, U. S. N.,
742 W. College Ave., State College Ave., State College, Pa.

Green, James Robert, Lieut., j.g., U. S. N.,
Electronics Materiel School, Treasure Island, San Francisco, Calif.

Green, W. H., Captain, U. S. N. R.,
1441 SW 11th St., Miami 35, Fla.

Haddell, James V., Lieut., U. S. N. R.,
3128 Oakwood Ave., Lynwood, Calif.

Hay, Richard A., Lieut Commander, U. S. N. R.,
Test Engineer, Glenn L. Martin Co., Electrical Laboratory Mail 716 East Belvedere Ave., Baltimore 12, Md.

Horner, S. W., Lieut., U. S. N.,
Care CNO Op-27, Navy Department, Washington, D. C.

Hosey, William R., Lieut Commander, U. S. N.,
515 8th St., Laurel, Miss.

Jensen, Jack E., Ensign, U. S. N. R.,
Jensen Machine Works, Sheridan, Wyoming.

Jerbert, Arthur H., Lieut., U. S. N. R.,
40 Walnut St., Newport, R. I.

Jeremiah, William E., Lieut Commander, U. S. N.,
Officers Club, 1400 New Hampshire Ave., N. W., Washington, D. C.

Johnson, Edward W., Lieut., U. S. N. R.,
Development Engineer, Engineer Board, War Dept.
Mail 4313 North 9th St., Arlington, Va.

Kin, Michael, Lieut., j.g., U. S. N. R.,
64 Orchard Drive, Clifton (Allwood), N. J.

King, Randolph W., Lieut., U. S. N.,
37 Huron Ave., Cambridge, Mass.

Kwart, Philip, Ensign, U. S. N.,
BOQ, Building 228, Naval School of Electronics, Treasure Island, San Francisco, Calif.

LaBarre, Carl A., Lieut. Commander, U. S. N.

Lis, Stanley J., Lieut., U. S. N. R.

McGuire, Donald E., Lieut., j.g., U. S. N.,
BOQ, Treasure Island, San Francisco, Calif.

McKee, Andrew I., Commodore, U. S. N.,
Electric Boat Co., Groton, Conn.

McPike, Howard, Lieut., j.g., U. S. N.,
421 North 3d St., Danville, Ky.

Menard, Bronley M., Lieut., U. S. N. R.,
2330 9th St., Bremerton, Wash.

Millard, Junius W., Captain, U. S. N. R.,
Chief, Mech. Sec., Research and Development Branch,
Military Planning Division, Office of Q. M. G.,
Washington, D. C.
Mail 3516 Valley Drive, Park Fairfax, Alexandria, Va.

Mountcastle, Albert E., Lieut Commander, U. S. N. R.,
Chief Clerk to Public Works Officer, N. O. B. Guam, M. I.
Mail N. O. B., Public Works Div., Box 38, Navy 926.
Care Fleet P. O., San Francisco, Calif.

Nichols, S. G., Captain, U. S. N.,
3131 O St., N. W., Washington, D. C.

Norwood, Gustav A., Lieut Commander, U. S. N.,
1107 Perimeter Road, Harvarclerneness Village,
Fort Devons, Mass.

Overton, Charles W., Lieut. Commander, U. S. N. R.

Pearse, Richard Lee, Lieut., j.g., U. S. N. R.,
M. E. Development Division, Aluminum Co. of America.
Mail 2124 Northland Ave., Lakewood 7, Ohio.

Ramsey, Lyle B., Lieut. Commander, U. S. N.,
707 Greythorne Road, Wynnewood, Pa.

Roberts, James W., Chief Machinist, U. S. N.

Rudisill, Dietz A., Lieut., U. S. N. R.,
414 South Main St., Lenoir, S. C.

Sheppler, Marcux A., Lieut., j.g., U. S. N. R.,
2722 North 16th St., Philadelphia, Pa.

Sims, Harry E., Lieut., U. S. N.

Slatton, James H., Lieut., j.g., U. S. N.,
M. E., Los Angeles Municipal Airport,
Mail 206 South Witmer St., Los Angeles 28, Calif.

Small, William A., Lieut. Commander, U. S. N., Ret.,
De Veaux School, Niagara Falls, N. Y.

Smith, Arthur C., Commander, U. S. N.,
Code 660, Bureau of Ships, Navy Department,
Washington, D. C.
Steger, Wilfred C., Lieut., j.g., U. S. N. R.,
618 Skitters St., Rentun, Wash.
Stevens, Ralph S., Jr., Lieut. Commander, U. S. N.,
Long Cove Road, Gales Ferry, Conn.
Stuart, Robert M., Lieut., U. S. N.,
Apt. 7, 26 South Battery, Charleston, S. C.
Thompson, Neil W., Ensign, U. S. N.,
724 W. 14th Ave., Spokane, Wash.
Tyler, A. F., Chief Carpenter, U. S. N.
Valentine, Kendall A., Lieut., U. S. N. R.,
Senior Engineer, Jackson & Moreland, Engineers, Boston,
Mass. Mail 777 East Ave., Walpole, Mass.
Vasalle, Bronald J., Lieut., U. S. N. R.,
Mech. Design Engineer, Norwood Machine Works, Inc.
Mail 183 Washington St., Norwood, Mass.
Wallace, Thomas G., Commander, U. S. N.,
Office of Asst. Secretary, Naval Material Division,
Navy Department, Washington, D. C.
Walsh, W. C., Jr., Lieut. Commander, U. S. N.
Wettergreen, Carl L., Lieut., U. S. N.
Yeomans, E. E., Captain, U. S. N.,
Office of Asst. Secretary of the Navy, Navy Dept.
Mail 1525 North Jefferson St., Arlington, Va.
Zink, Maurice W., Lieut., j.g., U. S. N.

CIVIL.

Batchelder, A. H., Manager, Petroleum Products Research Development and Technical Service, California Research Corp.,
P. O. Box 1627, Richmond, Calif.
Bush, Horace P.,
169 Fernbrook Ave., Wyncote, Pa.
Curran, Thomas M., Professor of N. A., Webb Institute of Naval Architecture, New York, N. Y.
Mail 50 Union St., Tarrytown, N. Y.
Demont, Albert M., Asst. Sect. Head, Administrative, Equipment Section, Aero & Marine Div., General Electric Co.
Mail 6 Irving Road, Scotia, N. Y.

Ellis, Louis C., Senior Marine Engineer, Code 5816, Bureau of Ships, Navy Department.
Mail 344 North Glebe Road, Arlington, Va.

Jucciarone, Nicholas T., Electrical Engineer (P5), Bureau of Ships, Navy Department.
Mail 1423 Allison St., N. W., Washington, D. C.

MacIntyre, David, Marine Development Engineer, Aluminum Company of America.
Mail 124 Argonne Drive, New Kensington, Pa.

Logan, Frank G., Vice President, Engineering, Leonard Ward Electric Co., Mt. Vernon, N. Y.
Mail Smallwood, N. Y.

McIntyre, William John, Supervisor Laison Sec., SPCC Catalog Division, Mechanicsville, Pa.
Mail 1917 Redina St., Harrisburg, Pa.

Miley, Frederick B., Estimator, Combustion Engineering Co., New York.
Mail 4319 Clarendon Road, Brooklyn 3, N. Y.

Nelson, Earl Pierce, Electrical Engineer, Ward Leonard Electric Co., Mt. Vernon, N. Y.
Mail 442 Espanade, Pelham Manor, N. Y.

Peters, Sidney A., N. A. (P6) Bureau of Ships, Navy Department.
Mail 4831 36th St., N. W., Washington 8, D. C.

Schmidt, Richard L., Application Engineer on Government Projects, General Electric Co., 806 15th St., N. W., Washington, D. C.

Stockton, C. Barclay, Sales Engineer, Ward Leonard Electric Co., Mt. Vernon, N. Y.
Mail 176 Amsterdam Ave., Hawthorne, N. Y.

Van Lear, G. Michael, Electrical, Bureau of Ships, Navy Department.
Mail 3334 Valley Drive, Alexandria, Va.

ASSOCIATE.

Bennett, Roy G., General Electric Co., 140 Federal St., Boston, Massachusetts.

Brandreth, Thomas T., Commander, R. N., 17 Macauley, Ludcombe Hill, Bath, England.

Hanlon, John F., Manager Marine & Electronics Division, Westinghouse Electric Corporation, 1625 K St., N. W., Washington, D. C.

Hardin, Freeman Fain, Tennessee Department of Employment Security, Route 3, Greenfield, Tenn.

Hucke, Herbert M., Manager, Aviation Radio Sales, RCA Victor Division.
Mail 2 Linden Ave.

LeRoy, Frank S., Government Sound Section, Radio Corp. of America.
Mail Box 63, Sellersville, Pa.

Kinsinger, D. W., Supervisor, Government Sales, Western Electric Co., 120 Broadway, New York, N. Y.

Transferred Associate to Civil.

Metcalf, Milton M., Head Engineer, Chief Draftsman, Mare Island Naval Shipyard.
Mail 114 Militia St., Vallejo, Calif.

It is with regret that the Society announces the death of MR. ROBERT E. POST.

MR. POST was serving as a member of the Council at the time of his death.